Merit Stands By Itself

IGNORING new things and improved products, especially those in one's own field, is certain to leave one behind-the-times. But the results are even worse when one becomes so critical that he fails to recognize merit and sweepingly condemns the whole plan in principle because insignificant details are open to criticism.

What does it cost to recognize merit in a competing design and to frankly praise it among close associates in your own organization? Too many of us are afraid to do so because we think that such action is a sign of weakness, an admission of inferiority. Attitudes affect our thinking; to self-justify our oral condemnation we warp our logic in order to arrive at the desired conclusion. Having developed such a mode of thinking we soon unwittingly begin to apply it toward the justification of our own errors.

Many companies hold "pow-wows" at which their sales and engineering executives gather around a sample of the latest creation of a competing company and over-indulge in criticizing it and finding faults. Everything from the nameplate to the main casting comes in for its share. Such over-indulgence leaves the mind in a stupor and befuddles all attempts to think honestly. I have witnessed many conclaves at which a competitor's product was roundly criticized without a single word of praise or commendation. But I have never known such inspection to be fruitful of anything constructive in the way of improving the design of the company's own product.

The first step in progress is the recognition of greater merit where it exists. If greater merit is not the aim, the achievement will not represent progress; it will be retrogression. Because design is largely a matter of compromises, sound unbiased judgment is of greatest importance in evaluating the relative merits of the various design possibilities. And it must be borne in mind that the merit of a design feature is in no way determined by the name of the manufacturer who incorporates that feature in his product. Because the leading manufacturer in a given field advertises a certain feature, that feature does not necessarily have merit. Nor does it follow that a feature has no merit because the most backward manufacturer uses it. The merit of an accomplishment is entirely foreign to the person who engineers it.

Merit stands by itself. It must be judged on fundamentals, not on the basis of personalities, competitive situations or politics. The most difficult tasks that confront the mind are those that deal with the evaluation of things or acts. It helps considerably to keep a clear mind by seeking commendable things rather than seeking reasons for condemnation in order to feed the ego. And instead of over-emphasizing and magnifying the effects of possible shortcomings in another person's design, there is more to be gained by a thorough study and analysis to find out how to improve further its worthwhile features, how to eliminate its weak points and how to profit from the evident experiences of its designer. The final design will then reflect the accumulated experiences gained in creating all previous designs of that type.

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Public search room at the patent department accommodates 235 searchers. Attendants bring to the searchers the records requested

PROPOSED PATENT LAWS

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THE Constitution, in Article 8, gives Congress the power "to promote the progress of science and useful arts, by securing for limited times to authors and inventors the exclusive right to their respective writings and discoveries." The emphasis should be placed on promoting the progress of science and useful arts. In other words, the real importance of the patent law is not to the inventor, as such, but is to the public as a whole. The purpose of the patent system is to benefit the public as a whole. The patent system, in order to be properly operated, should be such as to cause or encourage inventors to produce or evolve new devices, new processes and new machines.

In order to benefit the public, however, it is, of course, necessary that inventors do not keep secret what they have done, but disclose and publish their inventions so that they may eventually become a part of the common knowledge and for the common use of humanity. It is essential, therefore, in considering the patent system to keep constantly in mind the two matters, (1) invention itself (2) disclosure of the invention.

The experience in the last hundred and fifty years under the American patent system seems clearly to indicate that it has encouraged invention and in such a way as to cause almost all inventions to be disclosed to the public. Proposed changes in the patent laws, therefore, should be carefully examined to see whether they are likely to change this situation, since substantially all progress might stop if inventors were not encouraged to produce new things and disclose them to the public. Those who drafted the Constitution and those who have drafted the patent laws as well as those who have administered and adjudicated the patent laws seem to have had these considerations clearly in mind.

The public have been told that if anyone makes an invention and files a disclosure of it in the Patent Office he will be given the exclusive rights to that invention for a limited period. The impetus to invention and the consideration for the disclosure has been this exclusive

right. This exclusive right is sometimes referred to as a monopoly, and indeed, in one sense of the term it is a monopoly. It must be remembered that a monopoly of something which has come into public use and is already being made by many people, such as playing cards, for instance, stands on a very different footing from a monopoly of something which is new and about which the public knows nothing, such, for instance, as the telephone when it was invented and first brought before the public. In one instance a monopoly takes from the public something that the public already has. But the patent monopoly may withhold from the public for a limited time only something which the public has never had and would not ever have except as a consideration for the monopoly.

The anti-trust laws are drawn to prevent monopolies. Thus they are directed at activities which restrain trade. Much litigation has been conducted to abolish monopolies as such. The courts have established that in general it is not lawful to restrict the amount of a product to be produced or to restrict the territory of a trader or to restrict the price under which articles are sold. These seem to be reasonable and legitimate inhibitions with respect to the sort of business in which anyone is free to engage. In some litigation, however, the defendants have said that what they are doing is not unlawful because they are merely restricting trade within the exclusive right granted by a patent. In other words it has been held that since a patent is a monopoly itself monopolistic practices under it are not unlawful.

Question of Constitutionality

The Department of Justice, however, at hearings last year before the Temporary National Economic Committee specifically attacked the exclusive-right of patent and said that they believe the law should be changed so as to take away some part of the exclusive rights under a patent. There are many who believe that the Constitution should be literally interpreted and that Congress therefore, has the right to grant an exclusive right in an invention but has no right to make a patent grant which is only partially exclusive or which is only exclusive in some respects. Passing over the question of constitutionality, however, it may be desirable to consider the proposals of the Department of Justice to see whether their tendency may be to kill the goose which has laid the golden egg for our American civilization. The amendments proposed should not be made in the patent laws if their tendency may be to discourage invention, because certainly the discouragement of invention is not to the public advantage.

While much research is conducted by wealthy corporations and many inventions are made by employees of wealthy corporations which have ample funds and facilities for promoting inventions, the fact remains that there are still many independent inventors. The tendency of a corporation's inventors is merely to improve the product of the corporation. We must in general, therefore, look to independent and unattached inventors for many of the really new inventions on which new industries may be based. It is very important, therefore, that the patent system be kept in a condition to encourage such independent inventors. Ordinarily an inventor must expend funds for experiment and advertising.

Proposed Restrictions

Especially when the invention is particularly new, large amounts of money must be expended to induce the public to accept the invention. It thus becomes necessary to make large investments, sometimes running into millions of dollars, before any return can be gotten. Under these circumstances those who make the investment must be assured that when the business gets going they will be able to sell the product at a price high enough not only to pay running expenses but also to compensate for the initial investment. Since such an investment is highly speculative it must be possible to offer larger than usual returns on the investment. It seems clear that this can only be done by granting to the inventor and those associated with him under the patent an exclusive right to control the industry. Without this right, a competitor could come in after the industry has been developed, and sell at a price only slightly above the cost of production, without regard to the original development investment. This would entirely take away all incentive to spend anything for development. Nevertheless, the Department of Justice says:

"It should be made unlawful for any person to sell or assign a patent, or to grant any right or license under a patent, on any condition which restricts the assignee or grantee in respect of the *amount* of any article which he may produce under the patent, the price at which he may sell any such article, the *purpose* for which or manner in which he may use the patent or any article produced thereunder, or the *geographical area* within which he may produce or sell such article."

These are absolute restrictions proposed and the Department of Justice says that they should be:

". . . supplemented by a further prohibition against any other restriction embodied in a condition to any such assignment or license, which would tend sub-

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Patents that have been granted for more than 2,140,000 inventions are kept in the file room. The number of copies is more than 100,000,000, the patent copies sold to the public being drawn from this stock

stantially to lessen competition or to create a monopoly, unless such restriction is necessary to promote the progress of science and useful arts.

"The owner of a patent would enjoy the full patent monopoly if he elected to retain the exclusive privilege of producing or selling under the patent himself. He would be free to assign the patent; to grant an exclusive license; and to grant licenses to anyone he pleased. But, if he grants a license, the license must be general and unrestricted, unless he is prepared to demonstrate that a particular restriction (other than restrictions in respect of price, production, use, or geographical area) is necessary to promote science and useful arts. Restrictions in respect of *price*, *production*, *use* or geographical areas would be unconditionally outlawed."

Such an attempt to limit the inventor's right to exclude would tend to make it impossible for an inventor and his associates to obtain proper compensation for the invention or a proper return for their investment and this would inevitably tend to discourage invention.

Licensing Restrictions

It is frequently almost essential for an inventor to grant licenses to several people. It may be that the invention may be useful in a dozen different industries. It may be useful, for instance, on automobiles, on airplanes, on railways, on boats and in stationary power plants. These are separate industries and to require that any license that the patentee might issue must permit the purchaser of the license to manufacture for all of these industries will certainly tend to limit the market for the patent so much that probably no license will be granted.

If the inventor can license separate individuals for separate uses he can vary his royalty in different industries in such a way as to insure production in all industries and thus better give the invention to all parts of the public who can use it. The manufacturer of one type of device might well hesitate to accept a license when he knows that a similar general license given to someone now in another industry must be broad enough to allow and possibly induce the other to compete with him. Such a restriction certainly is against the interest of the independent inventor much more than against the interests of the large corporation since the latter may be entirely satisfied to reap the benefit of the invention in its own industry and keep the invention from other industries, while it may be necessary for the independent inventor to get tribute from a number of industries in order to obtain adequate compensation.

A striking illustration that shows how unrestricted licenses would react to withhold from the public the benefits of the progress of science would be the invention of a highly improved type of internal combustion engine. Assuming that such an engine were invented by a small independent automobile manufacturer, and that a manufacturing license could not be granted other than one covering all industries and applications, it would be to the advantage of the patent owner not to license anyone. Only in this way could he be sure to prevent some one from using the patent in the manufacture of automobile engines and thereby rob him of the distinct advantage he had originally obtained by virtue of his invention. Such being the case, the invention would be automatically restricted to automobile engines and for the whole life of the patent it could not be applied to airplane, marine, and industrial engines unless the automobile manufacturer entered into these fields, which he probably would not do. The detriment to the public welfare is obvious.

The granting of licenses with price restrictions has been approved many times by the courts. It may become necessary for the patent owner who feels he cannot adequately supply the entire market, to grant licenses to others to aid him in supplying the market. But in order to assure the continuance of his own business under the patent it may be essential to insist that his licensee does not undersell him. The patentee in order to retrieve his initial development investment may find it essential to keep his price high. If he cannot cause his licensee to maintain the same price he is likely to refuse to license someone else to compete with him. Thus the inhibition against price restrictions may actually be to the detriment of the public by limiting the output under the patent and so depriving many members of the public from its benefits.

Territorial Limitations

Territorial limitations in licenses may be very important in order to procure a wide distribution of the invention. A patentee manufacturing in Massachusetts may find that in order to distribute in the middle west and on the Pacific Coast he has to tax the public with such high transportation charges that it is not beneficial to the public. He may find that he can conduct his business more satisfactorily and more economically and hence better from the public standpoint if he limits his own production and selling activities to a certain territory and allows someone else already established in another territory to manufacture and sell there. This seems to be a matter of business expedience which the patentee is better able to determine than anyone else. If a limitation is put in the patent law so that the patentee may not make this business judgment for himself the tendency is certainly going to be to discourage invention.

The patentee may have limited facilities for manufacture. When he finds another with ample facilities for additional manufacture who is anxious to take a license he certainly should be allowed to grant a license in such a way as to reserve sufficient business to keep his own business going. In many instances this can be done only by limiting the amount of production by the licensee. On the other hand, the patentee may hesitate to grant a license to a manufacturer unless that manufacturer agrees to make a certain quantity so as to provide an adequate royalty return to the patentee.

Nullify Intent of Patents

All of the restrictions which the Department of Justice proposes are real limitations on the right of the patentee to use his monopoly in any way he deems best to compensate him for his invention and development work. Any such restrictions certainly will make an inventor hesitate before making an invention. Then if an invention is made, such restrictions on the right to grant licenses will tend to cause the inventor to keep his invention, if possible, as a trade secret, and thus annul the very purpose of the patent system, which is to avoid trade secrets and procure publication and dissemination of knowledge to be used by the public at the end of the limited monopoly term.

Those favoring the restrictions proposed by the Department of Justice point to some of the testimony presented at hearings before the Temporary National Economic Committee. It is a matter of record that some of the testimony revealed at least one situation that might be interpreted as an abuse of the patent privilege. This was only one isolated case whereas much of the other testimony, particularly that given by executives of the automotive industry, showed clearly that in the main, American industry adheres pretty well to the spirit of the patent law as well as the letter of it. And, as pointed out by numerous authorities, in and out of the Patent Department, possible breaches of the spirit of the patent system will be largely eliminated by the speeding up of the operations of the Patent Department and the new laws restricting the life of the patent from its date of application.

[In the second part of this article, which will appear next month, Mr. Fenning discusses the proposed legislation designed to minimize Patent Office delays and to expedite court actions on patent suits.]

DIE CAST OR SAND CAST-I

HERBERT CHASE

BOTH sand castings and die-castings are products of wide and unquestioned utility. Both enter in a large way into the production of machine elements and many other products because of the economy with which they are turned out. In certain classes of products, neither can take the place of the other, but in the field of light and medium to small size castings, there is a measure of competition between the two. The choice of which to select depends upon many factors.

Much may be gained by an effort to evaluate the two types of product. It must be admitted, however, that no hard and fast rules can be laid down which will, in themselves, settle the whole question. Too many variables are involved and they change with market conditions, labor supply and the like. Some general statements may be made to help clarify the situation, but it should be understood that they are generalities and thus are subject to exceptions. Most of the important considerations are outlined in Table II and will be further explained. Finally, there are appended specific examples of actual experience based either on cases in which substantially the same part was made by die casting and by sand casting, with costs each way, or in which the part was figured for production in both forms and one form selected either on a price basis or for some other controlling reason.

Naturally, the author does not claim to be infallible, or even that his judgment may not be found open to criticism by others with more information or differing points of view. An effort has been made, however, to present in an impartial way, such facts as it has proved possible to secure in numerous contacts with those who are face to face with the problems as a part of their regular duties.

For simplicity, the comparison is based, except where otherwise indicated, on unalloyed gray iron for the sand casting and zinc alloy for the die-casting, as each is most widely used in its respective field. Both are at the low end of the cost scale and both have good casting qualities. It should be noted, however, that where qualities not to be had in gray iron, on one hand, and zinc alloy, on the other are required, many other materials for sand casting and die casting, respectively, are available. Many alloy irons and numerous non-ferrous sand casting alloys are in extensive use and can match or exceed the physical and some other properties of the die-casting. For die casting, besides the zinc alloys, there are the aluminum and the magnesium alloys characterized by light weight; the brass alloys, some of which have physical properties comparable to mild wrought steel; and the tin and lead alloys for a few rather special purposes. There are instances where permanent mold castings, as distinct from the above types, deserve consideration and may be preferable to either a sand casting or a die-casting, but space for more than a mention of this alternative is lacking here.

Among the outstanding advantages of gray iron sand castings (to be dealt with in more detail under specific headings in subsequent paragraphs) are: low cost of material having many excellent qualities, moderate cost for patterns and tooling, ease of casting at a fairly good production rate, good machining properties, strength adequate for a wide variety of uses, and hardness and freedom from cold flow are often highly advantageous. These are offset, as compared with die-castings,

TABLE I—Physical Properties of Sand and Die-Castings

Material	Tensile Strength Lb. per Sq. In.	Compressive Strength Lb. per Sq. In.	Impact Strength Charpy Ft. Lb. ¼ In. Square Unnotched	Impact Strength Charpy Ft. Lb. 0.394 In. Square Unnotched	Shearing Strength Lb. per Sq. In.	Elongation in 2 In. Per Cent	Brinell Hardness	Spec. Gravity	Wt. per Cu. In.	Section Thickness Usual Min. In.	Commercial Dimensional Tolerances Inches per In.
Unalloyed gray iron sand cast	a 20,000–32,000	70,000 200,000 d	no data	2.1–2.9 c	25,000–80,000 d	Almost nil	100– e 700 f	7.00 g 7.70 h	0.252 g 0.278 h	⅛–¼ s	±1/32 s t
Zinc alloy die cast	40,300–47,900	60,500–93,100	18–20	no data	30,900–45,800	3.0–5.0	74–83	6.6–6.7	0.24	0.025 s	±0.001 k
Aluminum alloy die cast	b 32,000–34,000	no data	1.0–3.2	no data	18,000–26,000	1.0–4.0	60–95	2.66–2.98	0.095–0.103	0.030 s	±0.002 k
Magnesium alloy die cast	b 26,000–30,000	no data	1.0–2.0	no data	no data	1.0–3.0	60–62	1.81	0.066	0.030 s	±0.002 k
Brass die cast	b 55,000 95,000	no data	33 min.	no data	50,000–55,000 j	10–20	120–180	8.15–8.47	0.299–0.303	0.050 s	±0.003 k

(a) Ordinary gray iron. Physical properties can be greatly improved by alloying with nickel, chromium, etc.
(b) Same or similar alloys can be sand cast with same or, on some scores, superior physical properties. Wider choice of alloys available for sand casting.
(c) Values to 5.0 ft. lb. obtained with certain alloy iron, but note that area of specimen is 0.155 sq. in. or nearly 2.5 times that of die cast specimens (0.0625 sq. in.)
(d) Higher value is for alloy cast iron.
(e) Full annealed.
(f) Chilled and heat treated.
(g) Coarse grain.
(h) White iron.
(j) Yellow brass; probably higher in other brasses.
(k) Where controlled by integral die parts; somewhat wider as between moving die parts.
(s) For small castings.
(t) Wider where core work is involved.

by such shortcomings as: higher cost of machining, need for heavier sections and for wider dimensional tolerances, high cost of coring, low ductility and low impact strength.

Included in the advantages of die-castings are: rapid production within dimensions so close that little machining is needed, thin sections and lighter weight, accurate coring at only slight or no extra cost, smooth surfaces without sand inclusions, lower finishing costs and superior strength, especially in impact. Offsetting these are such items as: higher material cost, higher cost for dies than for patterns, certain limitations on shape, especially in some types of coring and, in most alloys, softer materials, some of the latter being subject to cold flow.

Because of die costs, die-castings are seldom economical when quantities required run below 1,000 to 5,000 unless the corresponding and casting necessitates much core work, too much machining or too high costs for finishing. There are, of course, many large sand castings not capable of duplication in die casting, but there are also many small and some large die-castings quite impossible to duplicate in sand cast form on an economical basis.

Such are some of the generalities which are, for the most part, fairly well understood in the metal working industries. It is desirable, however, to set down the facts more fully under specific headings if one is to gain a more thorough insight into the subject.

MATERIALS AVAILABLE have been mentioned, and data on their respective properties are given in Table I. Relative to the whole group of die casting alloys, however, it may be said here that aluminum and magnesium alloys, and especially those copper alloys grouped under the term "brass," are harder to cast and involve greater die expense than the zinc alloys and those with still lower melting points. Besides being low in cost, zinc alloys can be readily cast with smooth surfaces, good properties, thin sections and minimum die costs. Consequently, other alloys are employed only when they yield particular properties not to be matched in the zinc alloys. There is no precise parallel for this in sand casting, but the low cost of gray iron and its fluidity in molten form, which makes for easy casting, as well as certain excellent properties, account for its preponderant use in sand casting.

SPECIFIC GRAVITIES of zinc alloy and gray iron are nearly the same: 6.7 average for zinc alloy and 7.28 average for cast iron. The slight advantage in total weight of the zinc alloy on this score is increased in many castings, however, by reason of the thinner sections in which it can be cast and still have a wide advantage over cast

TABLE II—Comparison of Factors Affecting Choice of Casting

	DIE-CASTINGS	SAND CASTINGS
MATERIALS	Zinc alloys lowest in cost and most widely used. Aluminum alloys next in cost and extent of use. Alloys of copper (brass), magnesium, tin and lead for special purposes	Gray iron lowest in cost and widely used. Alloy cast iron available for many special purposes. Numerous non-ferrous alloys available for special purposes
WEIGHT (*specific gravity*)	Zinc alloys average 6.7. Aluminum alloys average 2.75. Magnesium alloys average 1.81 (lightest), lead alloys average 10.65 (heaviest)	Cast iron average specific gravity about 7.28. Range of gravity in non-ferrous alloys is the same
SECTION THICKNESS	Ranges from 0.025 in. upwards in very small castings, from 0.040 in. upwards in medium size castings and from $\frac{1}{16}$ in. upwards in large size castings	$\frac{1}{16}$ in. minimum attained in few specialized castings, but $\frac{3}{32}$ to $\frac{1}{8}$ is considered usual limit in small castings with $\frac{3}{16}$ to $\frac{1}{4}$ in. minimum in larger sizes
CORING	Can be complex, cores as small as $\frac{1}{32}$ in. diam. can be used. Fine threads can be cast where saving over tapping results. Cores involving undercuts are not practical. Effect of coring on casting cost slight	Can be complex and often involve undercuts but cores less than $\frac{1}{4}$ in. thick are seldom practical. Making and placing of cores greatly increases casting costs
EXTERNAL FORM	Can be complex and involve fine details but undercuts often require extra die parts	Can have almost any shape and include undercuts but these may greatly increase molding costs
SURFACE SMOOTHNESS	Can be such that only buffing is needed to insure excellent surface for plating	Usually requires grinding, polishing and buffing or equivalent to a good degree of smoothness
DIMENSIONAL ACCURACY	Commercial tolerance ± 0.001 to 0.003 in. per in. on most important dimensions	Commercial tolerance $\pm \frac{1}{32}$ in. for important dimensions on small castings, but is increased for core work
TOOLING (*including die for die casting*)	Die cost higher than pattern cost but tooling for machining is likely to be lower because of fewer operations and lighter cuts	Pattern costs may be 15 to 25 per cent or more of cost of die. Tooling for machining likely to exceed that for die-casting
PRODUCTION RATE	50 to 1,000 cycles an hr. (average about 200 to 300 an hr.) one to several castings per cycle (one man), time for core operation included	25 average small molds an hr. (one man) not including pouring, or making or placing of cores. One to several castings per mold
MATERIAL COST (*melting extra*)	Average about $8\frac{1}{4}$ c. per lb. for zinc alloys (but castings often weigh less than sand castings)	About $1\frac{1}{2}$ c. per lb. plus cost of sand and core materials
COST OF MACHINING	Low because of close dimensions and smooth surfaces. Only light cuts when any are required. Many holes cored to size. Metals free from hard spots and sand inclusions and easily machined	High where close dimensions and smooth surfaces are needed; not hard to machine in general, but hard spots and sand inclusions tend to dull tools
COST FOR APPLIED FINISH	Low because of smoother surfaces (especially important where plating is required). Hidden parts require no finish as metal is not subject to red rust	Usually higher for equal smoothness and general appearance. Hidden parts often require finishing to avoid red rusting when in ferrous alloy
FASTENING AT ASSEMBLY AND INSERTS	Can have integral rivets or extensions to be headed or spun over, or accurate cored holes. Inserts readily cast in place	Integral fastenings usually not feasible and cored holes are relatively inaccurate. Inserts seldom feasible
STRENGTH AND DUCTILITY	Zinc alloys generally superior in tensile strength; much superior in impact strength and in ductility to gray iron (see Table I). Some other die casting alloys inferior to certain sand cast alloys	Gray cast iron superior in compressive strength but inferior in tensile and much inferior in impact strength and ductility. Some other alloys equal to or better than die casting alloys
HARDNESS AND COLD FLOW	Zinc and aluminum alloys range from 60 to 83 Brinell. Brass 105 to 180. Zinc alloys are subject to cold flow	Gray cast irons range from 100 to 700 Brinell. Dimensional stability of cast iron an asset in some applications but material is not entirely free from cold flow. (See A.S.T.M. Symposium on Cast Iron)
APPEARANCE	Excellent in as-cast form because of smoothness and color and in finished form where fine detail and sharp lines are required	Rough and, in some instances, lacks detail. Color of gray iron is dull

iron in impact strength. If aluminum and magnesium die-castings be compared with cast iron, the weight saving is still greater. (See Table I for relative specific gravities). The cost per die-casting in aluminum alloy is only a little higher than in zinc alloy, and in certain unusual cases, quotations are the same or slightly lower for aluminum alloy.

SECTION THICKNESS invariably has a lower allowable minimum in the die-casting than in the sand casting, and such thin sections even enable the die-casting to compete with stamping in numerous instances. Clock cases, for example, are produced with sections averaging about 0.030 in., and housings as large as 11 x 7 x 7 in. are made in sections averaging about 0.040 to 0.045 in. using zinc alloy. Aluminum die-castings do not have to be much thicker. As against this, although some special flat lock parts are said to be sand cast in iron only $\frac{1}{16}$ in. thick, $\frac{3}{32}$ to $\frac{1}{8}$ in. is considered low for small iron castings, and $\frac{3}{16}$ to $\frac{1}{4}$ in. is low for large iron castings. Even large die-castings such as full size automobile radiator grilles seldom exceed $\frac{1}{8}$ in. in average thickness, and often have thinner sections. Die-castings can have sections of $\frac{1}{2}$ in. or thicker, but are seldom economical except perhaps in small castings. Much heavier sand castings are produced, of course, and porosity may occur in thick sections in both die and sand castings.

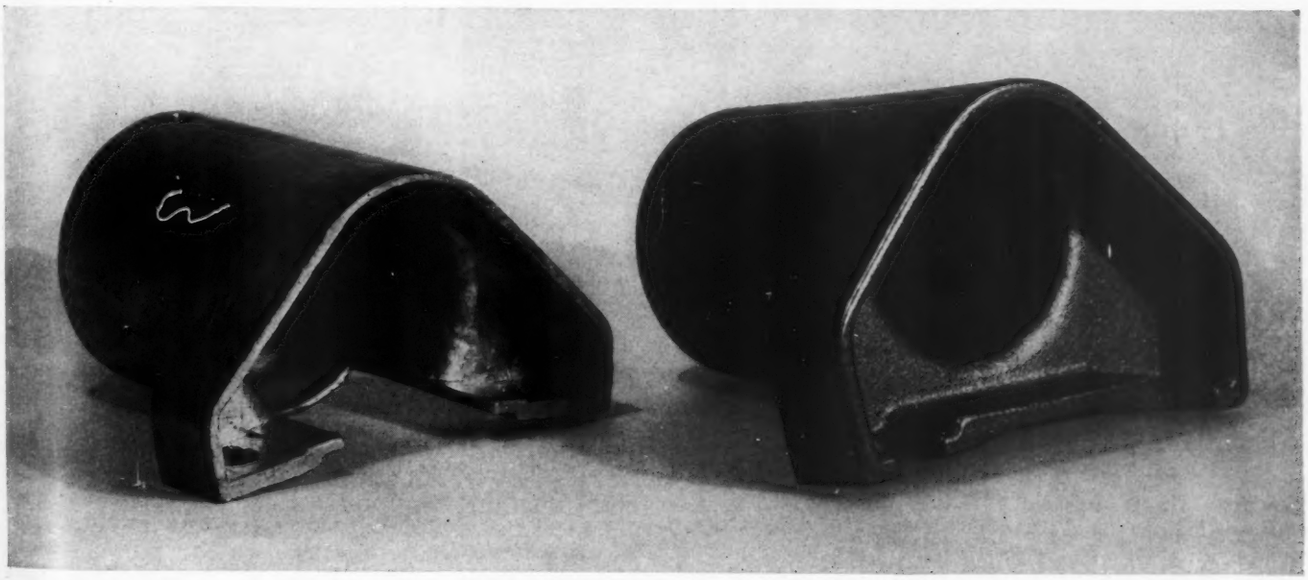
CORING involves several considerations which are different in die casting than in sand casting. In sand casting, cores as small as $\frac{1}{4}$ in. diam. are considered small and drilling usually preferable. Cores down to $\frac{3}{32}$ in. diam. are used in die-castings and almost any size hole, in which the depth does not exceed two to three times its diameter, can be and usually is cored, especially if in a favorable position. In general, however, the core cannot involve an undercut and, if it comes at an angle to the direction of die motion, some increase in die cost is involved. Sand cores can have almost any shape, but they have to be made up for each casting in special core boxes and involve several extra operations in production, and the placing of them in molds, slowing production accordingly. Operation of movable cores in die casting sometimes lengthens the casting cycle, but cores are often arranged for automatic operation and then have little or no effect on the time of the casting cycle. Even when coring increases die costs and lengthens the cycle, it often effects important economies by savings in machining and in metal. It may do likewise in sand casting, but the effect on cost is likely to be less favorable because of the extra labor and delays which cores entail. The fact that close dimensions are difficult to hold in sand coring may require added machining operations.

EXTERNAL SHAPES can often be the same in die cast as in sand cast parts, but in each type some shapes can be made which are not feasible in the other type. Since a die is not flexible, the die-casting must clear it when ejected, whereas the sand mold is destroyed after each casting. Patterns must clear the sand mold, of course, but they can be split or made with loose parts, or the mold can be parted in such a way as to yield castings not feasible in die casting. Many dies have slides carrying parts which form undercuts and the die can, in effect, be split in various ways when any extra costs entailed are justified. Greater faithfulness in detail and greater sharpness of lines and edges are feasible in die-casting.

SURFACE SMOOTHNESS invariably favors the die-casting or can be made to do so by care in finishing die surfaces. Sand moldings necessarily show the irregularities of the sand surface and tumbling and sand blasting do not remove these entirely. Die-castings are now produced with surfaces so smooth that only light buffing is required before plating.

DIMENSIONAL ACCURACY, being much greater in the die-casting, is among its important advantages. Relative commercial tolerances are given in the Table I, and the much closer limits laid in die casting tend to reduce machining costs to an important degree. Sometimes no machining, except to remove flash, is required. When machining is needed in the die-casting, less allowance for metal removal is made than in the sand casting.

The actual case histories presented here are based on parts which are or have been in production in die cast and/or sand cast form. Data in the examples given support the foregoing conclusions. The examples are instructive in themselves, as they constitute concrete evidence on costs and design advantages in a wide variety of applications.

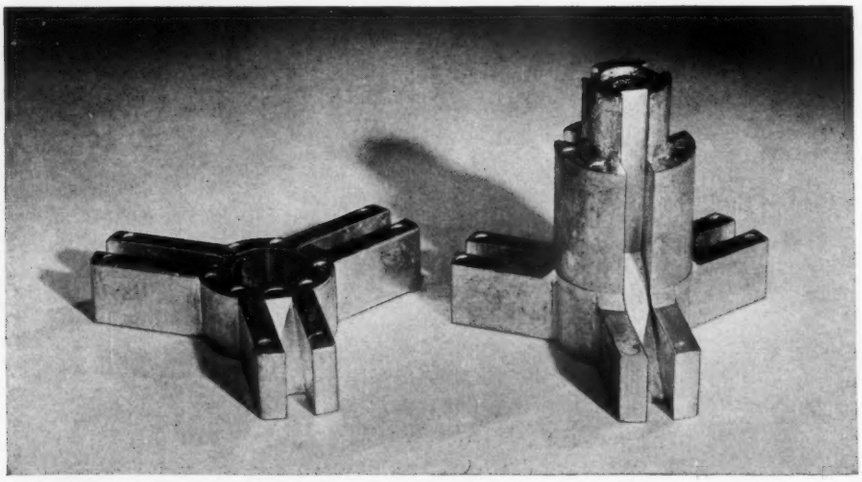


Simple castings having D-shaped recessed ends and tubular extensions. The parts have been produced in both die cast and sand cast form, and although not exact duplicates serve the same purpose, acting as brackets for supporting the vertical tube or "post" of a washing machine and attaching same to the tub. In the sand casting the tubular portion is somewhat longer than in the die-casting. The die-casting weighs about half as much as the sand casting. Some machine work was required on both, but the lower cost of material in the sand casting effected a net saving of 40 per cent in the total material and direct labor costs. A consideration of this cost advantage resulted in selection of the sand casting.

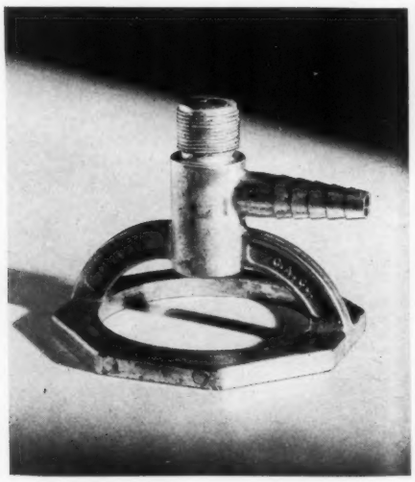
Paint tank cover subjected to air pressure, has been produced in both die cast and sand cast form; cost figures given here are based on lots of 10,000. Die-castings were in zinc alloy and were produced in a single-cavity die which, with tools for fin removal, cost $400. The price per piece was 28c. and was based on a production rate of 100 per hr. Machining cost 3c. per piece. For the sand casting, the pattern and match plate used in molding cost $75 and was made in two weeks, whereas the casting die required eight weeks to produce. The piece price for sand castings (from a two-impression match plate) is 17c., but machining costs 10c. per piece. Painting costs were 3c. per piece for both die-casting and sand casting. Advantages that are cited in favor of the die-casting are: less machining, better valve seats and fewer screw machine parts. Although the sand casting weighs 2 lb. 8 oz., which is one third more than the die-casting, the sand casting is now being used. The reason given is that the die-casting (the average section thickness of which is about 20 per cent thinner than that of the sand casting, and which appears to have been designed with inadequate stiffness) gave trouble from leakage around the seat because it bent or warped under the air pressure applied in service. The extra screw machine parts evidently increased the cost of the sand cast assembly, but specific data as to how much the cost was increased are lacking.

Barrel plug with 2 in. pipe thread, produced in both sand cast and die cast form. Die-casting die has 6 cavities, required 4 weeks to build, and cost $865 plus $74 for fin removal tools. Cast thread necessitates a die parting along length of piece. Die-castings in zinc alloy weigh about 5¼ oz., are produced 5,000 in 8 hr. at a price of 3½c. each.

Match plate for gray iron sand castings has 6 impressions, cost $150 and required 6 days to make. 700 sand castings, without thread, weighing 7 oz. each, are produced in 8 hr. at a cost of 3c. each. Machining costs 3½c. per piece, about 1 oz. being removed. Die-castings do not rust, they do not require cadmium plating when used in barrels containing alcohol.



Cylinder ridge reamer body castings produced in die cast form, a zinc alloy being used, in a die which cost $250. The castings have three radial arms which are slotted, the slots having the same width as splines in hub portions. The projecting splines in the hole of the smaller casting mate with corresponding spline recesses in the hub of the larger piece, which has an O.D. of 2½ in. over ends of arms and a height of 2 in., the hub being 1 in. O.D. necked to ⅝ in. O.D. to fit the hole in the smaller part. These castings cost 12c. per pair in lots of 2,000 and the only machining needed is the tapping of the central hole in the larger piece at a cost of $\frac{3}{16}$c. per piece. It has been estimated that sand castings could be produced at about 3c. per pair from a match plate costing about $100, but they would require so much machining to realize the close fits attained without machining in the die-casting, that sand castings are not considered feasible.



Bunsen burner base, 3¼ in. O.D. and height 2⅜ in., was originally a gray iron sand casting made from patterns costing $50. Sand castings cost 12c. each and machine work was 10c. per piece additional. Part is now die cast in zinc alloy in a single-cavity die which cost $250. The piece price is 6½c. in lots of 2,000, no machining is required. The fine thread on the projection is cast on. Die-castings are finished in gray gunmetal lacquer, sand castings are finished in black japan.

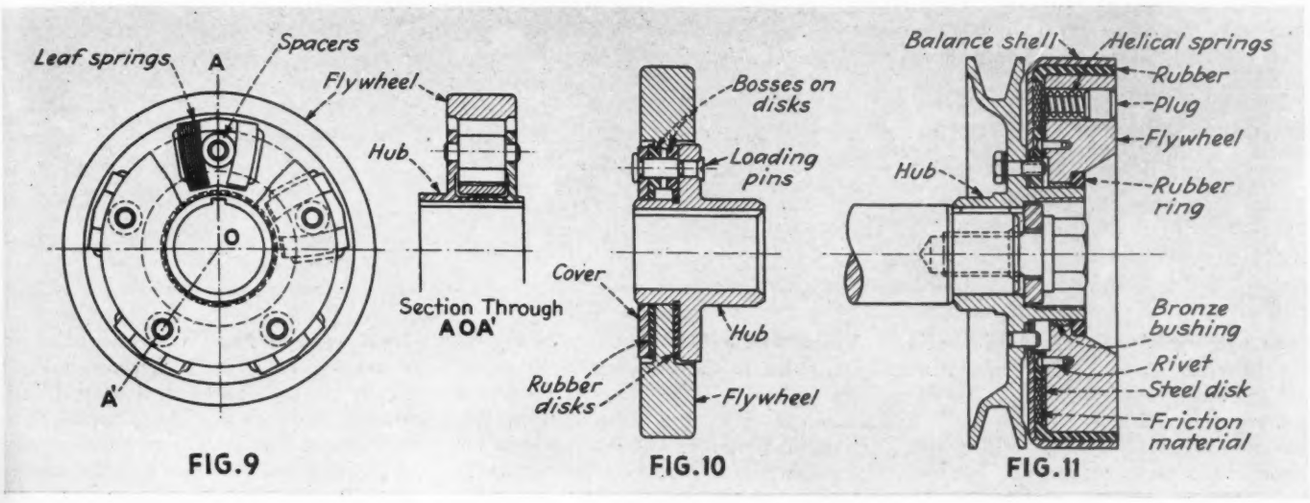
DAMPERS AND SUPPRESSORS–II

For Controlling Torsional Vibrations in Crankshafts

D. GERDAN
Detroit Diesel Engine Division of General Motors Corporation

TUNERS having no damping are limited in their application to constant speed engines or to those having a very narrow operating speed range. Although critical speed of the original system is eliminated with the application of the tuner damper, two new critical speeds arise, one is above and the other is below the original. Unless damping is introduced into the tuner, these new criticals often result in excessive amplitudes. Consequently, for wide speed range engines, tuners incorporating damping are a necessity. These consist of a flywheel mass connected elastically to the crankshaft through leaf springs, or rubber having hysteresis qualities. The action is the same as that in the tuners except that the energy causing the excessive torsional vibration at the two new critical speeds is dissipated by means of the inter-leaf friction or the rubber hysteresis, and amplitudes at these criticals are held to safe values.

Tuners With Incorporated Damping Elements



Suppressor shown in Fig. 9 is similar to that shown in Fig. 6 with the exception that flat leaf springs instead of the helical type are used as the elastic element. The use of packs of leaf springs introduces damping and makes it possible to suppress torsional vibrations over a wide speed range instead of only at one speed as with a straight tuner. The added damping is derived from inter-leaf motion in the spring packs.

This design consists of a flywheel connected elastically to the hub through several packs of leaf springs. The packs of springs are located and loaded through the spacers which are fixed in the hub. The hub is keyed to the crankshaft. The spring packs are given an initial deflection so that they are not loaded at any time during operation.

The unit as used on passenger car engines is not lubricated, being mounted on the front end of the crankshaft in the open. Some designs are inclosed within the crankcase at the front end of the engine and provision is made for lubrication of the spring packs either through low pressure or by splash feed.

Rubber elastic elements used in a damped tuner rather than springs. In the design shown in Fig. 10 the flywheel is connected elastically through two rubber disks to the hub which is fastened to the engine crankshaft. The two rubber disks have raised bosses, molded on one side, that when assembled fit into holes in the web of the flywheel. Loading pins pass through the bosses on the rubber disks. These loading pins also serve to hold the assembly together. A shoulder on the pins seats against the hub and serves to hold the cover at a definite distance from the hub thus giving a fixed amount of compression of the rubber disks.

The raised bosses in the rubber disks guide the balancer flywheel in relation to the center of the assembly. They also constitute the portions of the disk which act as the elastic members. When the flywheel oscillates relative to the hub, the rubber is compressed against the pins and the holes in the flywheel. Damping is obtained from the sliding of the flywheel surfaces on the thin portion of the rubber disks.

Damped tuner in which flywheel mass is coupled elastically to the hub by means of rubber bonded both to the flywheel and to the balancer shell is shown in Fig. 11. The shell is fastened to the hub which is keyed to the crankshaft. Friction, in addition to the small amount obtained from hysteresis of the rubber, is obtained by use of friction material. The friction surface is a ring fastened to a thin steel disk riveted to the flywheel. The friction surface is loaded by springs having a fixed initial compression set up by the plug pressed into the flywheel. The flywheel is guided on the hub by a bronze bushing.

In operation, the flywheel oscillates relative to the hub and because of the elasticity of the rubber elements counteracts the exciting torques with the vibratory torque of the flywheel. Damping is also obtained to further reduce the torsional vibration from the sliding of the friction material upon the balancer shell. The combination of optimum tuning plus damping effects gives a balancer effective through a wide engine speed range. (*Continued on next page*)

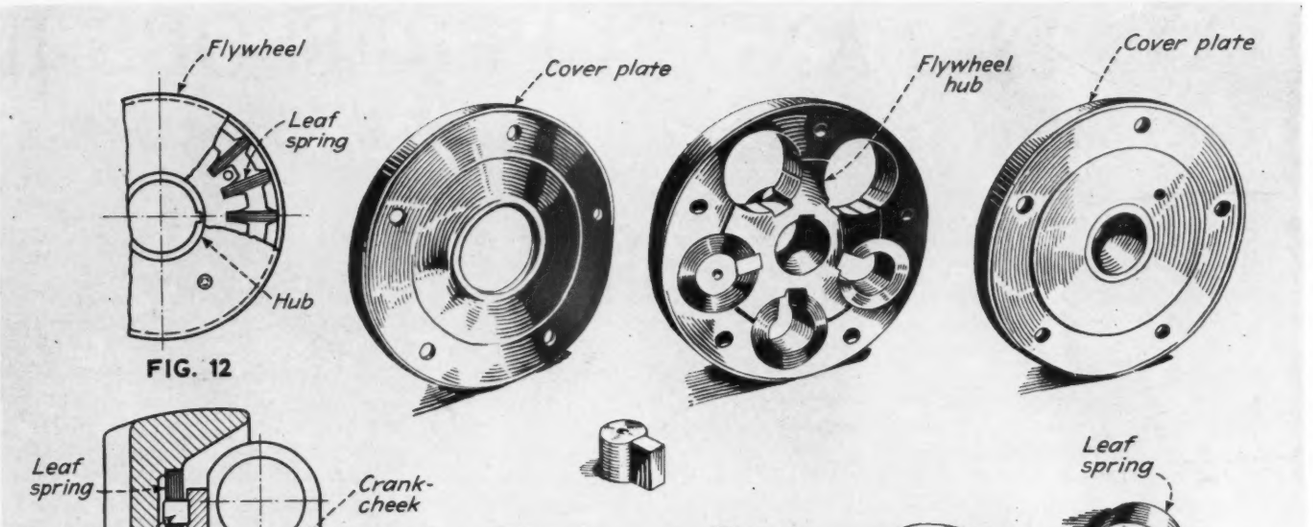


FIG. 12

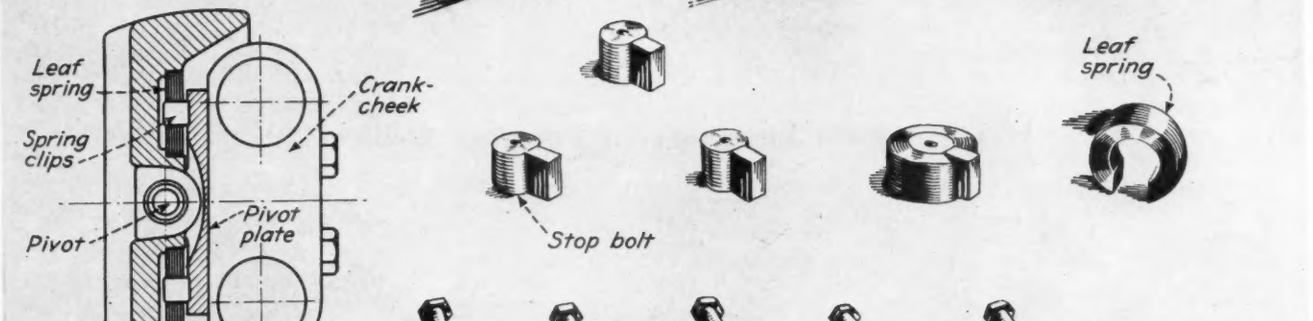


FIG. 13

FIG. 14

Leaf springs in the design shown in Fig. 12 replace the helical springs of the straight tuner shown in Fig. 6. The flywheel is mounted elastically on the hub through the leaf springs which are pressed into slots cut in the flywheel and the hub. The hub is keyed to the crankshaft. This type can be made to have a variable natural frequency by proportioning the sides of the spring slots to cause a decrease in the effective length of the spring pack, thus giving increased rate of action to the packs. Damping effects are introduced through interleaf friction caused by the deflection of the spring leaves when the flywheel oscillates relative to the hub. This design can also be used as a damped elastic coupling located between the engine and the equipment driven by it.

Vibration balancer shown in Fig. 13 is similar to the one that is shown in Fig. 8. The helical springs of the design shown in Fig. 6 are replaced in this vibration balancer by leaf springs for the purpose of introducing damping qualities in the system.

The inertia mass is pivoted on a pin carried by the plate which is fastened to a crank cheek at the front end of the crankshaft. The leaf spring packs are held in clips. The outer ends of the leaf spring packs bear against pockets in the inertia mass. The spring clips are fastened to the pivot plate as shown. The spring packs constitute the elastic and damping elements. The springs are given an initial deflection so that complete unloading will not occur even at the maximum oscillation amplitudes.

Leaf spring type vibration balancer consisting of a flywheel coupled elastically through packs of C-shaped leaf springs to a hub fixed to the crankshaft is shown in Fig. 14. The thickness of the spring material decreases with the diameter. The springs, when assembled into the unit, fit around bolts which carry a lug which acts as a stop, preventing the spring pack from turning in the bore formed by the flywheel and hub. The bolts also serve as a stress limiting device. The assembly is held together by cover plates fastened to the flywheel thus adding to its inertia.

As the flywheel oscillates relative to the hub the packs close slowly on the stop bolts, increasing the spring stiffness and results in a balancer having a variable natural frequency.

De-Tuners That Vary the Frequency of the Crankshaft System

These devices automatically vary the natural frequency of the crankshaft system thus preventing the build-up of resonant conditions. They consist essentially of a flywheel connected to the crankshaft either flexibly or through friction. If connected flexibly the springs are such that their rate of action varies, causing the de-tuner to have no fixed natural frequency of vibration but rather a series of natural frequencies dependent upon the deflections of the springs.

The frictionally connected types closely resemble dampers and in effect may be considered as such in that a heavy flywheel mass coupled to the crankshaft through friction is used. The flywheel rotates with the crankshaft through the influence of friction until a certain predetermined maximum angular acceleration caused by torsional vibration occurs, then slippage of the flywheel relative to the shaft takes place, and since the flywheel is in effect disconnected from the crankshaft system, the natural frequency of torsional vibration of the system is increased. The unit having very little hysteresis, functions in the same manner as pure tuners and is unsuitable for wide speed range operation. However, through suitable compounding of the rubber stock an appreciable amount of hysteresis can be incorporated in which event the unit will function as a damper tuner, thus increasing its effectiveness over the entire speed range of engine.

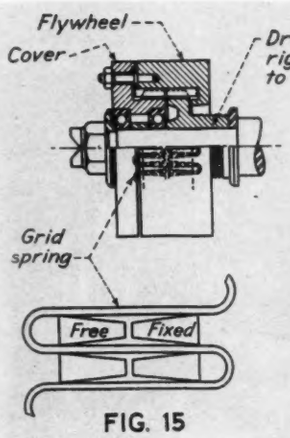


FIG. 15

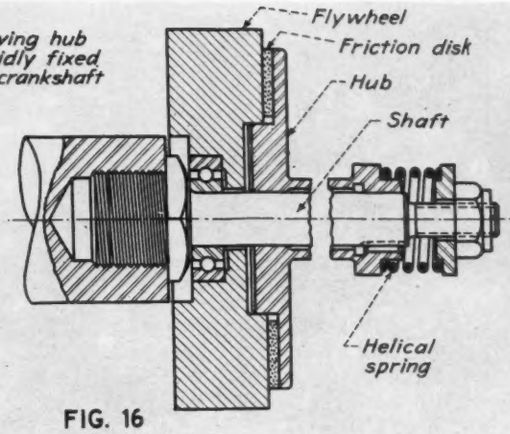


FIG. 16

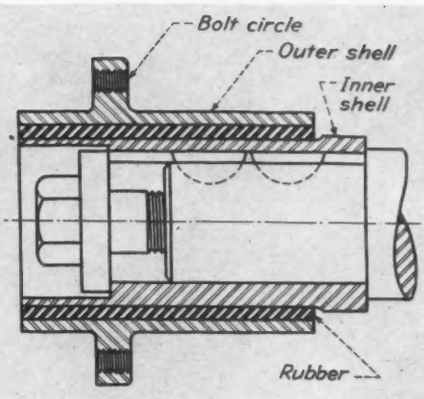


FIG. 17

De-tuning device which functions by preventing the building up of resonance conditions rather than by damping or absorbing crankshaft torsional vibrations. This unit shown in Fig. 15 consists of a flywheel coupled elastically through a special form of a "grid" spring to a hub which is fixed to the crankshaft. An automatic change in natural frequency of the system at critical speeds is obtained from the shortening of the spans of the springs as the increased torque is applied. Through this automatic change of natural frequency the amplitude of torsional vibration of the crankshaft at critical speeds will not attain the maximum otherwise obtained.

This design also finds application as a drive coupling between an engine and the driven equipment, in which application the cover which is shown here as part of the flywheel is disconnected and is connected to the driven equipment. The automatic change in frequency is obtained with the shortening of the spring length as the increased torque is applied near the critical speed and the vibration torque is prevented from reaching the minimum and maximum values.

Friction type de-tuner which may also be called a damper since it has all the characteristics of one is shown in Fig. 16. This design also has some of the characteristics of a damped tuner, and consists of a heavy flywheel coupled frictionally through a friction disk to a shaft of comparatively low torsional rigidity by means of a hub. The shaft is fastened to the front end of the crankshaft as shown. The shaft extension which has low torsional rigidity gives this design the characteristics of a damped tuner. If the shaft were rigid the unit would be a straight damper. The purpose of the flexible shaft as used in this device is to reduce the inertia, and therefore the weight, required in the heavy flywheel. The friction surface is loaded through the helical spring.

Slippage of the flywheel during operation in a critical period accomplishes two things. First, slippage gives rise to damping forces which dissipate the vibrating energy. Secondly, since the flywheel is, in effect, disconnected from the crankshaft during slippage, the natural frequency of the crankshaft system is automatically raised or increased.

Flexible driving hub. At times an engine will have no serious amplitudes of torsional vibration in its operating range, but with the addition of pulleys or fans directly connected to the front end of the crankshaft serious vibration occurs because of lowering of the natural frequency of the crankshaft system through the addition of these masses. The lowered natural frequency of the system results in a drop of the engine speed at which resonance occurs.

To avoid changing the position of the criticals in the speed range a flexibly mounted hub of the type shown in Fig. 17 can be used. This consists of a shell unit keyed to the crankshaft in which inner and outer shells are bonded together with rubber. The rubber as a rule has considerable elasticity. The member being driven directly from the crankshaft is then bolted to the outer shell which is isolated elastically by the rubber from the main crankshaft system.

Proper proportioning of the design will permit the addition of considerable mass to the front end of the crankshaft without an appreciable change in the crankshaft vibration characteristics.

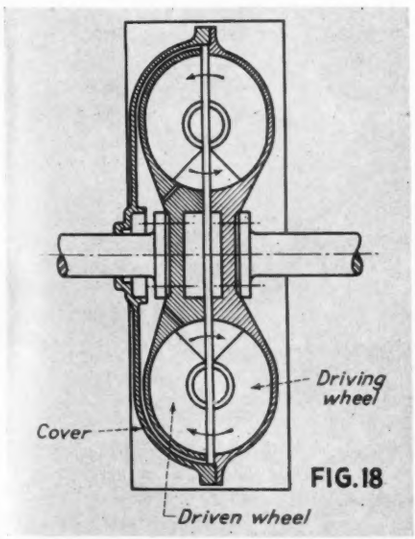


FIG. 18

Hydraulic coupling of the type shown in Fig. 18 consists of a cover bolted to and rotating with the driving wheel which is coupled to the engine. There is no connection between the driver and driven wheels other than oil carried in segmental cells formed in the wheels. The first effect of this coupling is to break up a relatively complicated vibrating system into its main components which prevents the transmission of strong torsional oscillations from the engine to the driven shaft, or vice versa.

With the driven member at rest and the driver starts to rotate, the oil carried in the cells of the driving member flows to the outer circumference under the action of centrifugal force and enters the cells of the driven wheel. In this wheel, still at rest, the fluid flows radially to the inner circumference and re-enters the driver, establishing a circulating motion of the oil. Since the direction of motion of the fluid is completely reversed in passing from the driver to the driven wheel, the torque reaction of the driven wheel must exactly equal the torque of the driving wheel.

Full slippage will occur as long as the driving torque is less than the driven torque. With an increase of driving wheel speed, the torque rises rapidly until soon it is sufficient to overcome the resisting torque and the driven wheel commences to rotate with consequent reduction of slippage.

At full speed the slip is about two to four per cent. This hydraulic slippage exerts a considerable damping effect upon such torsional vibrations as they tend to occur in both the driving and driven parts of the system.

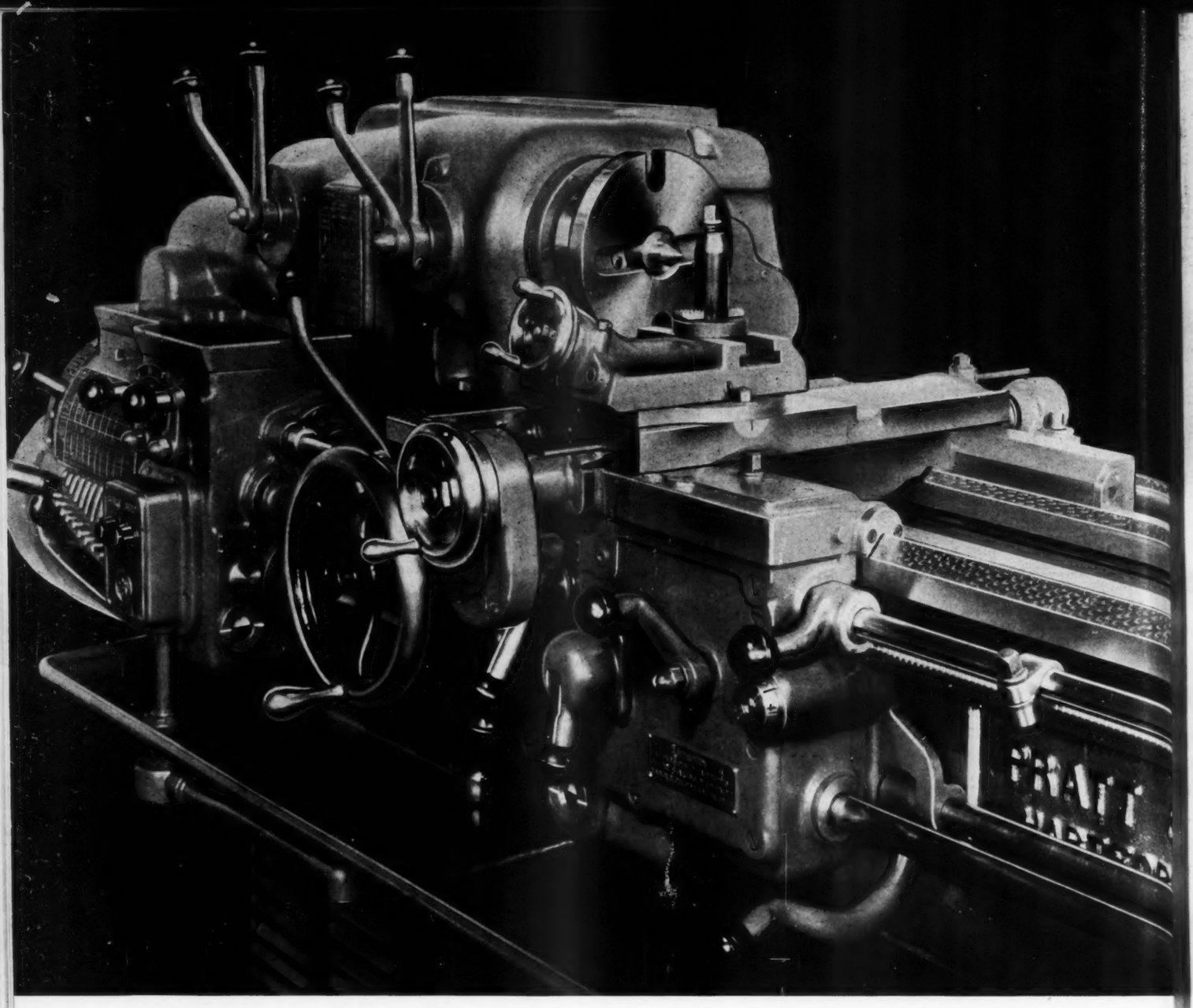


Fig. 1—Application of a completely concealed automatic centralized lubricating system to the carriage of a Pratt & Whitney lathe

AUTOMATIC LUBRICATION

WARREN O. WRIGHT

A CLOSE examination of latest models of machine tools, will reveal certain definite design trends with reference to lubrication. First, it becomes apparent that designers consider central lubrication an integral part of the modern machine tool. Secondly, where lubrication is required at many important surfaces, the system of central lubrication is usually selected as the best method of providing lubrication. And third, the various arrangements of oil-distribution and feed indicate the designers' extreme freedom in applying the principles of centralized lubrication to individual machine standards and requirements. A number of typical applications of automatic centralized lubrication as on a lathe but applicable to all types of machinery are shown here, giving an indication of the many possibilities when applying automatic centralized forced-feed lubricating systems.

The lubricating system as incorporated in the new Model C Pratt & Whitney lathe is a compact unit construction which, as shown in Fig. 1, has enabled the designer to eliminate most of the oil-feed tubing by providing suitable drilled passages in the machine parts.

Through careful design, and cooperation between the designer and the maker of the lubricating system, it was possible to automatically lubricate all bearings, gears and other surfaces on the apron and carriage at a minimum cost both for parts and assembly time. Moreover, doing away with tubing resulted in exceptional neatness of design.

Fig. 2—Function of Filler Rods. Passages (oil manifold) are drilled in the carriage as illustrated. As will be observed all drilled passages are partly filled with rods—these filler rods having a very important function.

Each operation of the lubricator introduces approximately 12 drops of oil into the system. This small quantity of oil must be distributed at 10 points. In the case of small-diameter tubing less than ⅛ in. bore, there is practically no tendency for air bubbles to be trapped. With drilled passages, however, it is found most economical, from a machining standpoint, to drill the longer holes of a larger diameter. This presents a possibility of air bubbles being trapped in the passages. As a safeguard against the presence of air in the system, suitable filler rods are used of a diameter about 1/32 in. less than the drilled hole. The elimination of air in the passages, combined with the use of a check valve at each outlet, maintain the distributing system full of oil at all times. In this way, even a small volume of oil, discharged by the pump, results in a definite pressure rise, so essential to efficient distribution. To properly distribute the oil, a pressure of 5 to 8 lb. per sq. in. is advisable. By use of filler rods, the volumetric capacity of the system is reduced to a point which results in a rise in pressure to approximately 40 to 50 lb. per sq. in. at each operation of the pump.

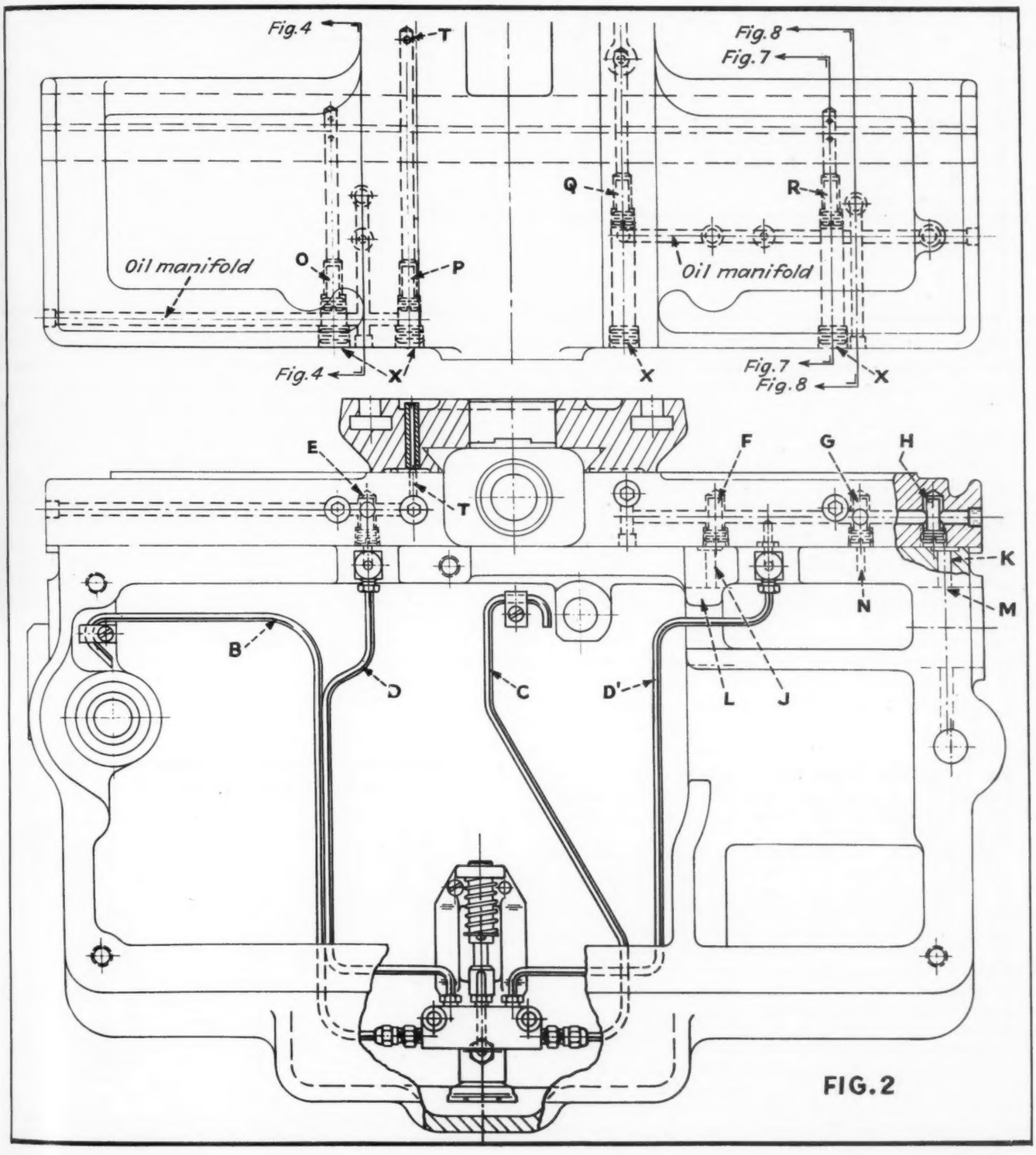


FIG. 2

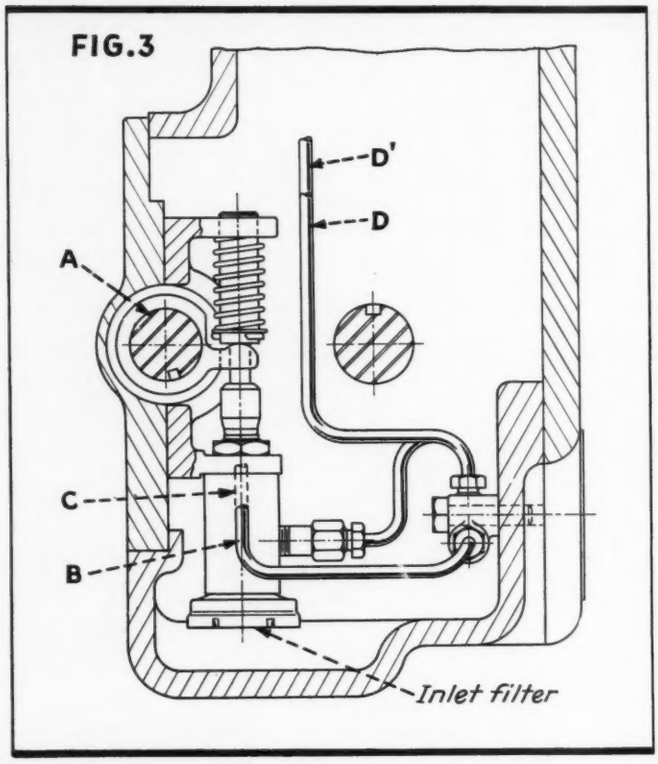


Fig. 3—Primary Oil Leads. The lubricator is mounted on the rear surface of the interior of the apron in such position as to assure the inlet filter being submerged in the oil. A slight depression under the lubricator inlet forms a pocket for sludge or other foreign matter. Shifter lever shaft *A* has a lug engaging the lubricator plunger. The rotating of this shaft, in starting the feed, raises the lubricator plunger; drawing a total of 12 drops of oil through the dense felt-filter into the pump cylinder. On reversal of the shaft motion, in stopping the feed, the plunger is lowered by the pump spring—thereby discharging the oil into the system under a pressure of approximately 50 lb. per sq. in. at normal room temperature. Volume of oil delivered is always constant regardless of temperature, oil viscosity or other factors, since a fixed volume pump is used. Tubes *B* and *C* feed oil to gear trains. This method was decided upon, although the tubes could have been dispensed with, as has been done at other points. Tubes *D* and *D'* are primary leads to the oil manifolds, as shown in Fig. 4, communicating with drilled passages in the carriage.

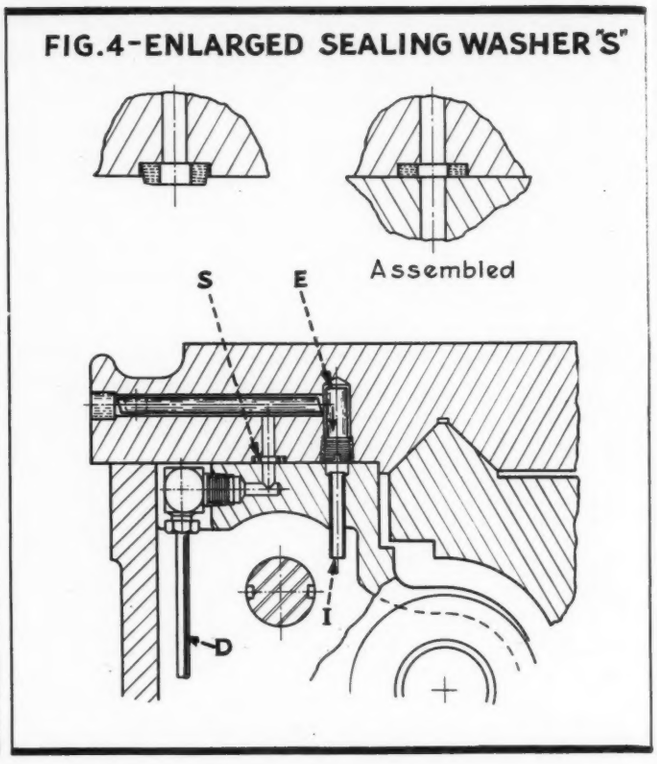


Fig. 4—Assembly for Oil-Tight Joint. A simple and effective means of connecting an oil passage to an adjacent machine part. The carriage is drilled and counterbored to carry a sealing washer *S* of synthetic material, as shown in enlarged detail. The sealing washer is cut with angular sides, the larger diameter being assembled in the bottom of the counterbore to hold it in position until assembly. When the carriage is assembled to the apron, the seal is compressed, making an oil-tight joint. This construction avoids the need for tubing connections, etc., when assembling and disassembling the units. Tube *D* is a primary lead to the oil manifold, communicating with the drilled passages in the carriage. The part *E* is the meter unit and *I* is the drip tube to provide lubrication for one of the gear trains. The horizontally drilled hole has a filler rod to decrease the volume.

As will be seen in the enlarged section, the washer in the free position provides a slight clearance on the top diameter, making certain that no material will be displaced and slip into the joint between the two adjacent parts when assembling.

Fig. 5—Special Meter-Units. Special meter-units are assembled in pockets communicating with the oil manifold, and largely fill these pockets—again to reduce the volumetric capacity of the system.

The special M type meter-units are similar in outer dimensions, but, by reversal of the inner parts and check valve, the oil feeds through the meter-unit in opposite directions in the two models; feeding towards the threaded end in the MKA type, and from the threaded end in the MKB. Either of the types is used, depending on the construction of the parts. In Figs. 2, 4 and 8, MKA meter-units are at *E*, *F*, *G* and *H*, and are installed from the bottom of the carriage. Meter-unit at *E* is mounted directly over a drip tube *I* (see Fig. 4), oiling one of the gear trains. Meter-units at *F* and *H* feed through passages *J* and *K* to the bearings *L* and *M*. Meter-unit at *G* supplies oil to the half nut *N*. Meter-units at *O*, *P*, *Q* and *R* are of the MKB type, and are assembled in pockets between the oil manifold and the bearing. Pipe plugs *X* (see Figs. 2 and 7) close the passages and permit removal of meter units if required.

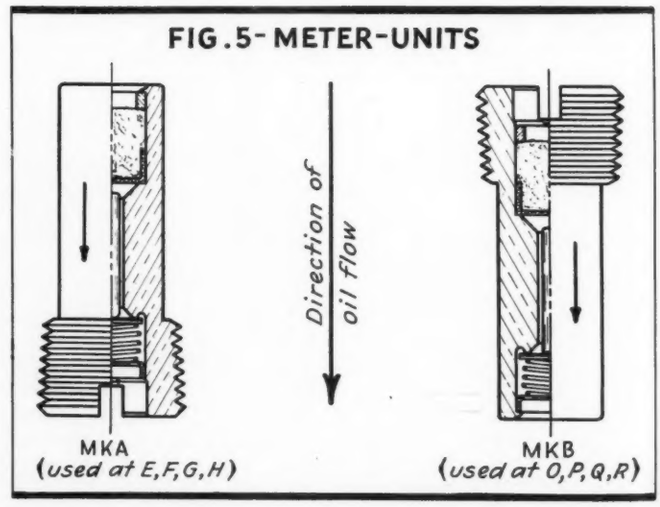


Fig. 6—Cross Slide Lubrication. Meter-unit at *P*, see Fig. 2, feeds the oil through a passage to the small hole *T*, which is located so as to communicate —at all positions of the cross slide— with the oil groove *U* shown here. A tube *V* is assembled in the cross slide— the lower tube end in the oil groove *U*, and its upper end slightly above the bottom of the oil reservoir *W*. Any oil flowing through the tube overflows into the reservoir, which is cast in the upper surface of the cross slide as shown.

At *Y* and *Y'* are assembled feeder tubes for the slides, and at *Z* is assembled a tube to feed the feed screw nut. These tubes are positioned so that their tops are at the same height, a little below the top of tube *V*. Oil overflowing from tube *V* raises the oil level in reservoir *W*, which overflows equally through tubes *Y*. *Y'*, and *Z*.

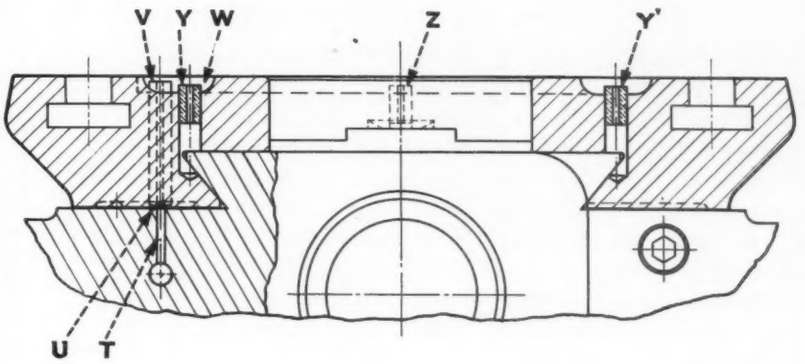


FIG.6

Fig. 7—Oiling the V Ways. The two sides of the V ways are lubricated from one point. Oil is fed to the passage *A* from meter-unit at *R*. Two holes are drilled from the top part of the bearing to the drilled passage. Being small in diameter, these two holes act as capillary holes; and, as the passage *A* is practically filled, oil is fed to the bearings only on operation of the pump—which introduces oil into the passage *A* under pressure, feeding about equally at both sides of the way.

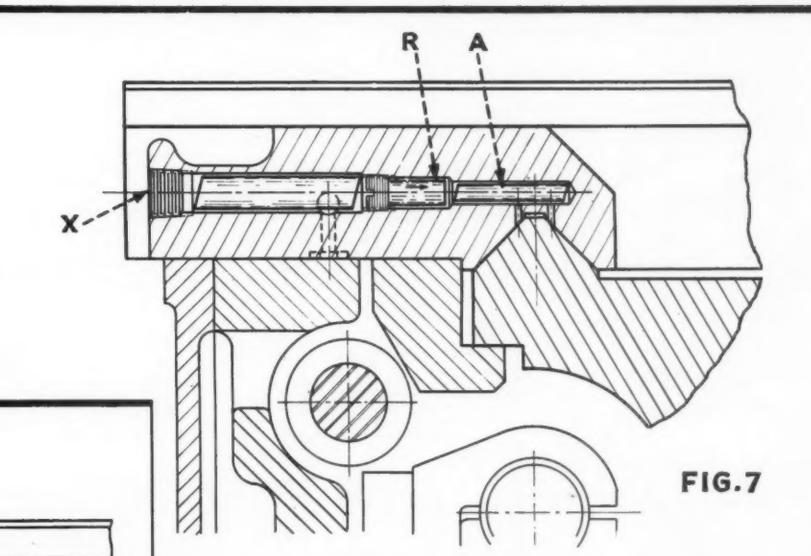


FIG.7

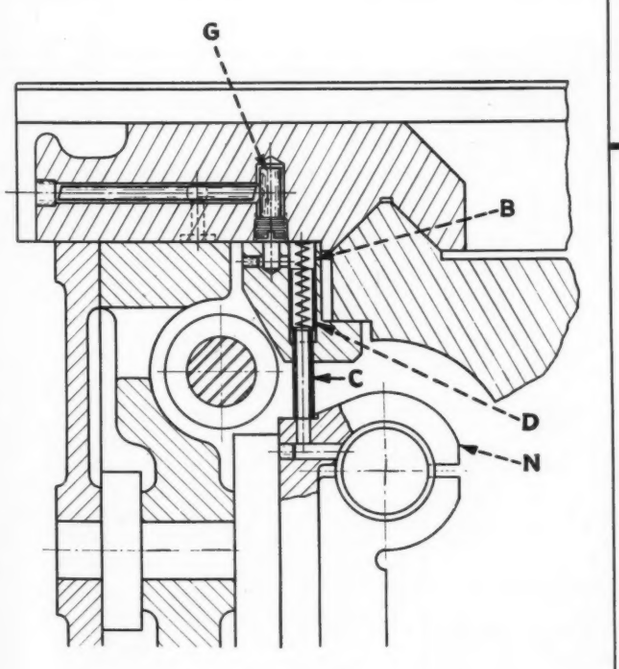


FIG.8

Fig. 8—Hollow Plunger Assembly. Oil from meter-unit *G* is carried through drilling to the hole *B*. In this hole is assembled a hollow plunger *C*, so mounted that the spring *D* maintains a constant bearing between the plunger end *C* and the half nut *N*, on movement of the half nut. A drilling in the half nut carries the oil to the thread. Because of the fit of plunger *C*, very little loss of oil occurs, especially as the oil at this point is not under pressure.

Simplicity of Arrangement. Careful study of the various diagrams presented will clearly show that, despite the ingenuity employed in the oil-feed to the different parts, the entire oil-distribution arrangement is kept simple. Operation of the lathe has consistently demonstrated the trustworthiness of its lubricating system—every bearing served, automatically receiving the required oiling.

A point which again may be emphasized, is that the designer could have varied the arrangement considerably and still could have retained maximum lubricating efficiency. Automatic centralized pressure-lubrication thus lends itself both to machine requirements and to the designer's preferences and skill in arrangement of the various units which make up the system.

SCRAPERS AND WIPERS

PROTECTION against damaging effects of small particles, which enter between ways, guides and similar sliding surfaces, is necessary to increase the accurate life of a machine. Most designers have provided this protection with brass, bronze, or cast iron scrapers which clean the ways without scratching; with felt, synthetic rubber or leather wipers which mop up oil, water and small particles; or with combinations of scrapers and wipers. Diagrammatic sketches on these pages illustrate adaptations of these devices to all types of sliding surfaces.

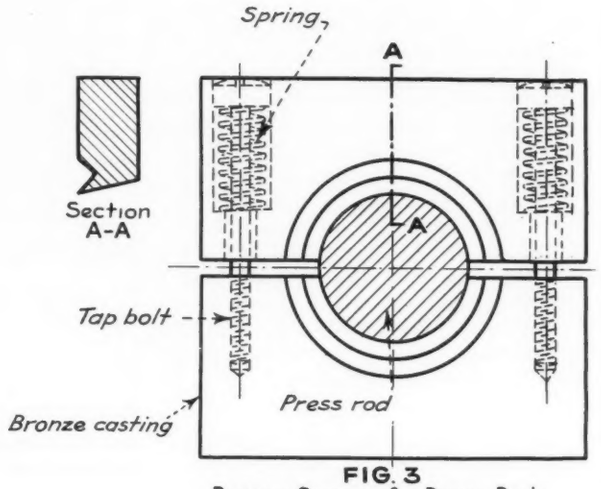
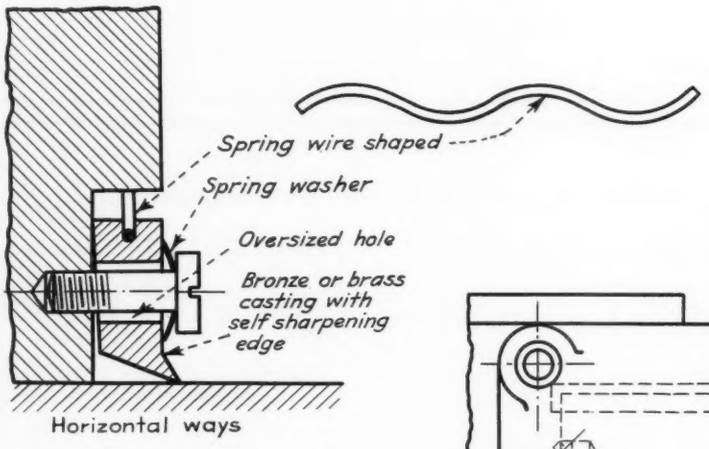


FIG. 3
Bronze Scraper for Press Rods



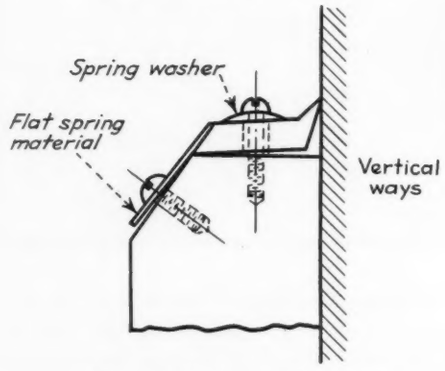


FIG. 1 Self-Sharpening Bronze or Brass Scraper (Patented)

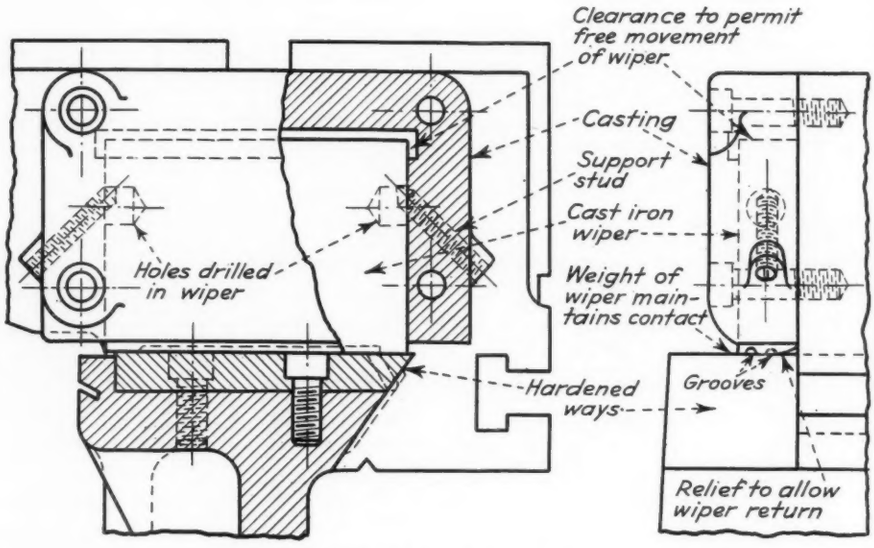


FIG. 4 Cast Iron Wiper

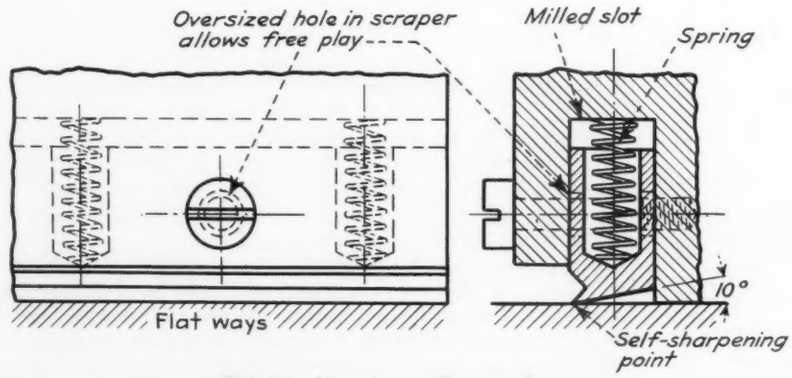


FIG. 2 Aluminum-Bronze Scraper

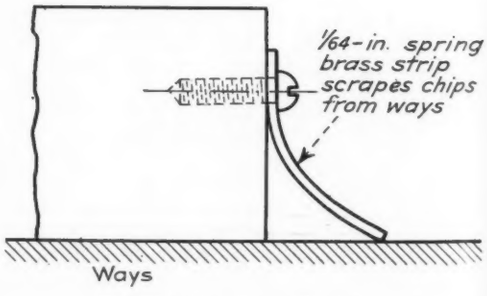
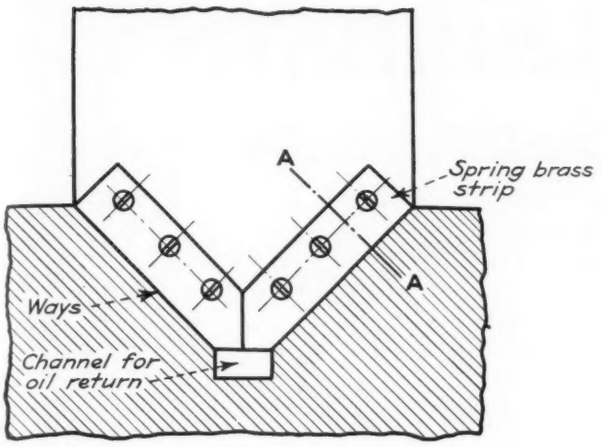


FIG. 5 Brass Strip



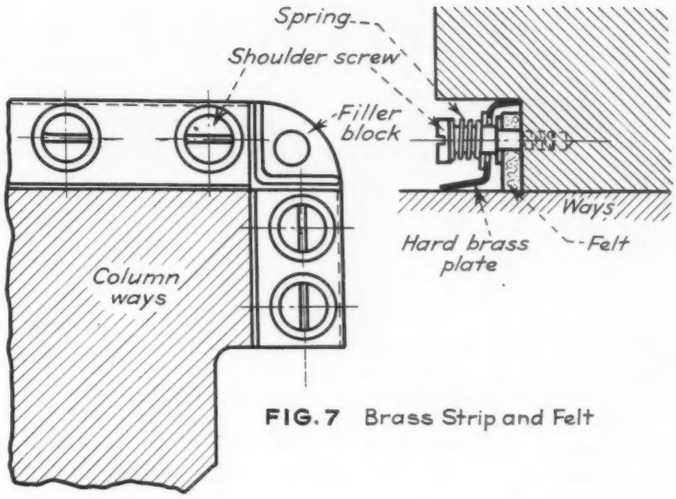


FIG. 7 Brass Strip and Felt

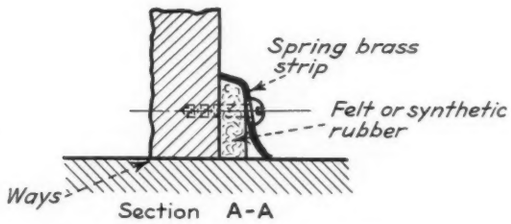


FIG. 6 Brass Strip and Felt

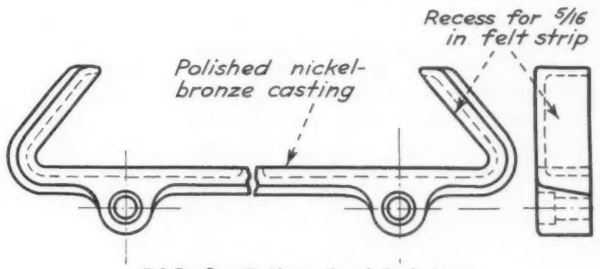


FIG. 8 Felt in Cast Retainer

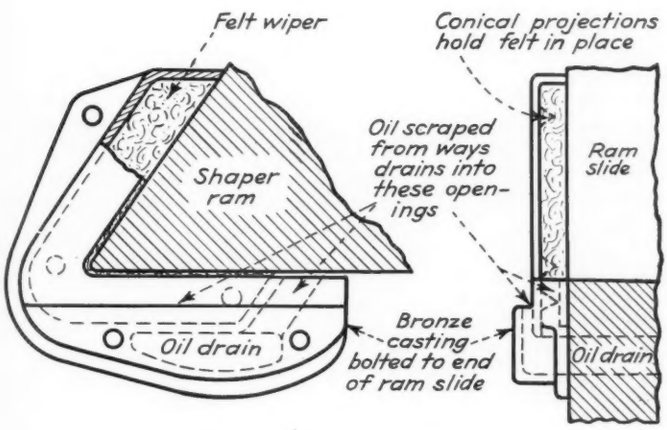

Felt wiper
Conical projections hold felt in place
Oil scraped from ways drains into these openings
Ram slide
Shaper ram
Oil drain
Bronze casting bolted to end of ram slide
Oil drain

FIG. 9 Felt in Bronze Retainer

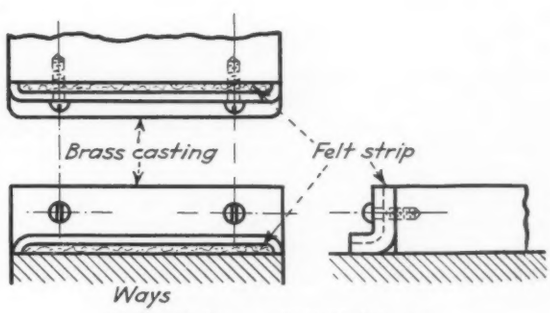


FIG. 10 Brass Felt-Retainer

SOME MACHINES are designed so that dirt and chips are kept entirely away from bearing surfaces. For example, leather aprons and spring expanding fabric bellows protect the ways of machines such as grinders and routers, while sheet metal guards are placed over the ways of other machines to deflect the chips. Telescoping brass tubes have been used to protect splined rotating shafts.

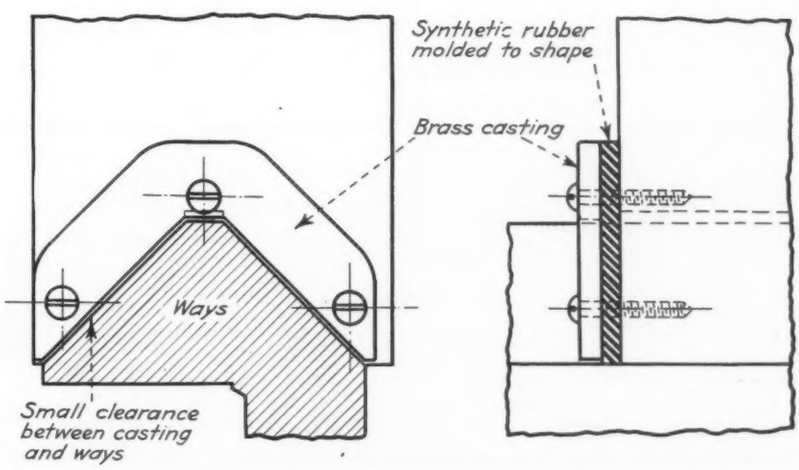


FIG. 11 Synthetic Rubber in Brass Retainer

BELLEVILLE SPRINGS

For Spring Thrust Loaded Spindles

WILLIAM WALLACE BOYD
Mechanical Engineer

SPRING loading of spindles has been limited in its application because of the large outside diameter required, and the length necessary for the spring and its deflection to obtain the requisite load. Also with helical springs there is the possibility of the spring unwinding or winding up with the rotating spindle.

A recent method for preloading ball bearings is to displace the inner race by a predetermined amount of thrust and grind this face flush with the side of the outer race. The bearings are interchangeable and an exact degree of tightness is always possible, but expansion or contraction can cause a change in tightness which will promote serious heating and cause excessive wear. It is customary to fix the main load carrying bearing in position and to provide a floating bearing on the opposite end of the spindle to accommodate changes in length.

If the main load carrying bearing is of the angular contact, ball or roller, type and a Belleville spring is on the floating end so that there is a thrust between the finished surface in the housing and the outer race of the rear bearing which will preload the main bearing, a rigid bearing is obtained in which there is not any radial or axial movement while under load, and this load will be elastic enough so that, over the usual range of difference in length between the spindle and its housing, there will not be any overloading.

Assume that the spring thrust load on the lathe spindle shown in the illustration is 12,000 lb. without any reference to the radial load, and that an angular contact ball bearing of approximately 4 in. I.D., $10\frac{7}{8}$ in. O.D., $2\frac{3}{8}$ in. wide, with 2 in. diam. balls, will be used for the main bearing; it is rated for an axial load of 22,000 lb. at 500 r.p.m., so that it will have a margin of 10,000 lb. over the spring load to take care of thrust in service machining operations.

The tail bearing will be of the angular contact type, approximately $3\frac{1}{2}$ in. I.D., $8\frac{7}{8}$ in. O.D., $2\frac{1}{8}$ in. wide, with $1\frac{1}{16}$ in. diam. balls. It has a thrust carrying capacity of 18,000 lb. at 500 r.p.m., all of which could be utilized, if it were necessary or desirable.

With the assumption of an axial working load of 12,000 lb., and that the working load is 2/3 the flat load, the flat load for the springs is 18,000 lb. Also assume that the O.D. of the spring is $8\frac{3}{4}$ in. and the I.D. is 4 in., giving 2.19 for the diameter ratio.

When P = total load, lb.
δ = deflection of each disk, in.
t = thickness of disk, in.
h = free height of each disk in.
α = ratio of O. D to I. D.
a = ½ outside diam., in.
E = 30,000,000 lb. per sq. in. for steel
S = stress at inside circumference, assumed to be 200,000 lb. per sq. in.
σ = Poisson's ratio, 0.3 for steel

$$M = \frac{6}{\pi \log_e \alpha}\left[\frac{\alpha - 1}{\log_e \alpha}\right]^2$$

$$C_1 = \frac{6}{\pi \log_e \alpha}\left[\frac{(\alpha - 1)}{\log_e \alpha} - 1\right]$$

$$C_2 = \frac{6}{\pi \log_e \alpha}\left[\frac{\alpha - 1}{2}\right]$$

From the *A.S.M.E. Transactions* of May 1936, page 305 (Almen and Laszlo), the formula for the load on each disk is found to be

$$P = \frac{E\,\delta}{(1 - \sigma^2)\,M\,\alpha^2}\left[(h - \delta)\,(h - \delta/2)t + t^3\right]$$

and the stress formula to be

$$S = \frac{E\delta}{(1 - \sigma^2)\,M\alpha^2}\left[C_1\left(h - \frac{\delta}{2}\right) + C_2\,t\right]$$

By calculating for the solid load first and making $\delta = h$, a thickness of 0.42 in. is obtained for each disk.

If a pile of four disks is used the working data will be:

Outside diam.	$8\frac{3}{4}$ in.
Inside diam.	4 in.
Diameter ratio	2.19
Working load	12,000 lb.
Flat load	18,000 lb.
Deflection per disk	0.42 in.
Thickness of disk	0.42 in.
Flat height	1.68 in.
Total deflection	1.68 in.
Free height	3.36 in.

To compare this Belleville spring with a helical spring of the same capacity and deflection,

Let P = solid load or 18,000 lb.
d = diam. of the wire, in.
S_s = allowable unit stress 80,000 lb. per sq. in.
r = pitch radius of the coil or $3\frac{1}{2}$ in.

$$P = \frac{\pi\,d^3\,S_s}{16\,r}$$

and substituting these values

$$d = 1.587 \text{ in.}$$

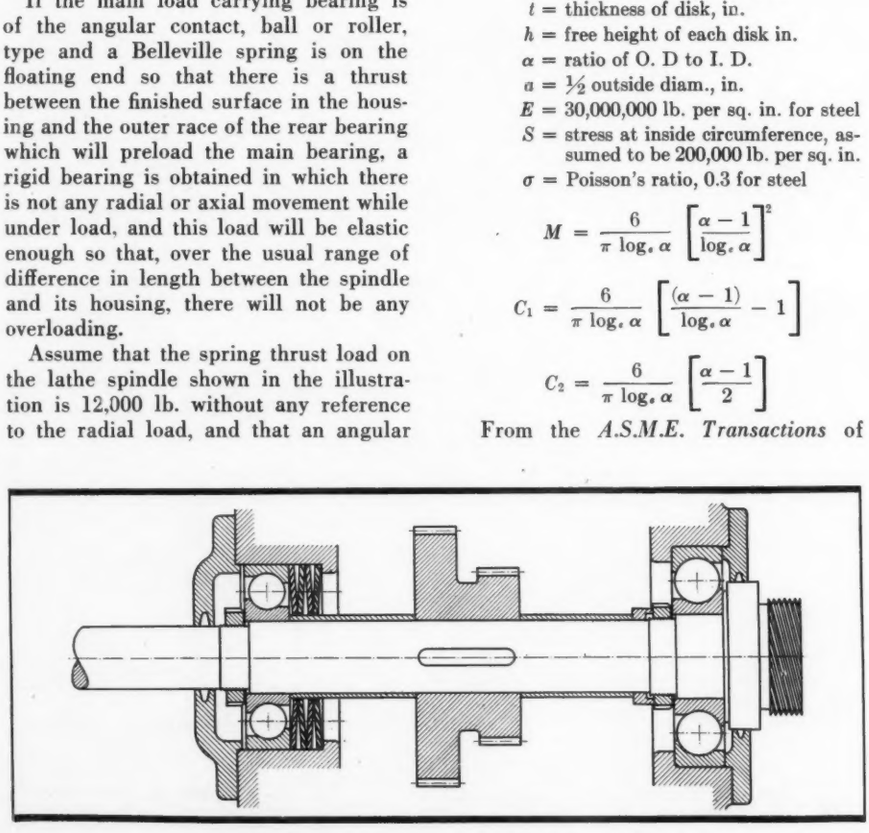
Taking d as $1\frac{5}{8}$ in. and G = 15,000,000 the deflection is found from

$$\delta = \frac{4\,r^2\,S_s}{d\,G}$$

to be 0.505 in. per coil.

The pitch of the coils then is 2.130 in. A deflection equal to that of the Belleville spring would require nearly $3\frac{1}{2}$ coils, making a total of about $7\frac{1}{2}$ in. without counting the dead coils nor the necessary spring seats.

From this comparison, it can be readily seen that the Belleville spring requires less space than a helical spring of equal capacity and deflection, and it is also much more economical to manufacture. There is an additional advantage, the Belleville spring resembles the elliptical spring in its softness of action.



Belleville spring thrust loaded lathe spindle

NICKEL ALLOY STEELS

Characteristics and Properties Required for Tractor Parts

H. L. GEIGER
Development and Research Division, The International Nickel Company, Inc.

WHEN it is considered that the power transmission groups which move tractors weighing from 2,500 to 32,000 lb. over all types of surfaces, practically none of which is smooth, and in addition must haul a load of two or three plows or disk harrows through all types of soil, operate a bull-dozer or haul heavy timber, it will be seen that no small engineering problem is involved in selection of materials and design. Therefore the particular alloys specified for the various parts of tractors are an excellent guide in the selection of materials for parts of other machines wherein comparable loads and operating conditions exist. Modern tractor design has not been so much concerned with reduction in weight as it has been in making a compact unit which meets the increasing demands for more power.

Singly or in combination factors such as resistance to wear, fatigue or shock, toughness, or high yield strength may be of prime importance in the design of a particular part. Many of the properties desired in the various tractor parts are indicated in Table II, and in Table IV the chemical analysis range of the steels referred to are presented.

The various shafts in tractors are designed principally for torsional loads and deflections rather than to resist bending, and because of the high torques involved the sections are relatively large.

Experience has shown that tractor shafts which have uniform cross-sectional hardness give least trouble in respect to torsional failures and permanent set under torque loads. The nickel, nickel-chromium, and nickel-chromium-molybdenum steels are known to respond to hardening in this manner throughout the relatively large sections involved in tractor design. The results obtained through raising the over-all hardness in quenching with the accompanying increase in strength has given these steels a position of importance among tractor builders.

Tractor power transmission shafting is made of both carburizing and oil-hardening alloy steels. Carburizing grades are usually specified for the spline shaft on which the sliding gears are mounted. With alloy steel shafts the maximum torsional values are obtained as well as maximum wear resistance on the carburized surfaces. Alloy is preferred to the plain carbon steel because of the resultant higher and deeper core hardness accompanied by greater strength and toughness. Alloy grade with 0.20 carbon is preferred to 0.15 carbon because it provides higher torsional strength.

Not only for the spline shaft but for all other shafts where loading is not overly high but where maximum wear resistance on part or all of the shaft is required, carburizing alloy grades are used. This practice is frequently followed where it is economical to forge a shaft and pinion integral. Typical applications on certain models are the power take-off shaft and pinion, belt pulley shaft and pinion, bull pinion and shaft; and, in some of the smaller models, axle countershaft and pinion. Four popular steels for these parts are S.A.E. 3120, 4620, 4820, and 2320, which by single or direct quench heat-treatment after carburizing will render the shaft or integral gears file hard or equal to a C 60-62 Rockwell hardness.

For parts where toughness and high torsional strength are prime considerations and resistance to wear is secondary, the so-called oil-hardening steels with their higher carbon content are satisfactory. The nickel steels suitable for shafting falling in this classification are S.A.E. 3140, 3240, 2345, and 4340, with alloy content increasing to offset "mass" effect in depth hardening. As a rule axles and shafts are treated to hardness ranging from 302 to 425 Brinell. (*Editor's Note:* The trend has been towards the higher hardnesses as it has been found that axles with a higher Brinell range from 400 to 450 have higher endurance limits, lessening the dangers of fatigue failures at the same time providing higher torsional strength.) The general practice is to harden the main portion of the shaft to a Brinell range of 320-350, and the ends which are splined to receive the wheel hub or differential gear, where failure is most likely to occur, are hardened variously from 365-425 Brinell, depending on the endurance requirements of the particular design.

The observations from torsion tests listed in Table I were made on full sized S.A.E. 3140 axles, water quenched and drawn to the indicated hardnesses.

The various specifications and values shown in Table III are for 3 in. diameter rounds; quenched and drawn at 1,000 deg. F. These values are presented primarily to show relative hardening and strengthening effects of alloy for a given diameter and draw temperature, using plain carbon 1040 as a basis for comparison. In Table III the effect of alloy content on "mass" is clearly shown.

In recent years the nickel-chromium-molybdenum steel, S.A.E. 4340, has been adopted for comparatively large diameter axles where the optimum in strength is desired. The higher alloy content of this specification produces a higher hardness on quenching and subsequent tempering than can be obtained in plain carbon or a lower alloy content steel. An outstanding feature of this steel is its response to machining at high hardnesses. Splines can be milled or hobbed in these axles at a hardness of about 450 Brinell under special conditions and at 388 to 411 Brinell under ordinary production conditions, with an 18-4-1 high-speed steel cutting tool and a sulphur-type cutting oil.

The 16 to 24 steel bearings including both ball and roller types in a tractor are mounted with great care. Mis-alignment of a shaft to 0.001 to 0.002 in. has been known to produce concentrated stresses on mating gear teeth of sufficient magnitude to produce failure. Mis-

TABLE I—Torsion Test Data

S. A. E. Spec.	Diameter at Point of Failure, Splined Ends	Surface Brinell Number	Torsional Elastic Limit In. Lb.	Torsional Yield Point In. Lb.	Angle of Twist at Rupture
3140.....	2½ in.	321	242,000	319,000	38 deg.
3140.....	2½ in.	352	275,000	357,000	34 deg.

TABLE II—Alloy Steels and Property Requirements for Crawler and Wheel Type Tractor Parts

Name of Part	High Yield Point in Tension	Resistance to Shock (Impact)	Impact Resistance at Low Temp.	Resistance to Wear	Resistance to Fatigue	Resistance to Pitting Fatigue	High Yield Point in Torsion	High Yield Point in Compression	High Torsional Impact	Toughness at Ordinary Temp.	Toughness at Elevated Temp.	High Yield Point in Bending	Resistance to Seizing	Ductility	High Yield Point in Shear	Depth Hardening in Heat-Treat.	Min. Distortion in Heat-Treat.	S. A. E. Alloy Steel Specifications Commonly Used (*0.15 and 0.20 carbon grades carburized; and 0.30 to 0.50 carbon oil hardened*)
Motor crankshaft	★			★	★		★		★	★			★	★		★	★	3140, X-3140
Piston pins		★		★	★			☆			★		★		★			3120, 4620, 2315
Intake valves				★	★						★		★				★	3140
Exhaust valves				★	★						★		★				★	21% Cr., 12% Ni.
Clutch hub and joints					★		★		★	★				★				3140
Clutch shaft					★		★		★	★				★				3140, X-3140
Transmission gears	★	★	★	★		★		★		★		★		★			★	3120, 4620, 2315, 2345, 4820
Transmission spline shaft			★	★	★		★		★	★			★		★		★	3120, 4620, 3140, 2315, 4820
Transmission countershaft					★		★		★	★								3140, 4340, 4820
Power take-off shaft	★		★		★		★		★	★		★				★		3140, 3120, 4620, 4320
Power take-off gears	★	★	★	★		★		★		★		★		★			★	3120, 4620, 4320
Belt pulley shaft	★		★		★		★		★	★		★				★		3140, 3115, 4620, 3120, 4320
Belt pulley gears	★	★	★	★		★		★		★		★		★			★	3120, 4620, 4320
Differential shaft				★	★		★		★	★			★		★		★	3120
Differential side gears	★	★		★		★		★		★		★		★			★	3120, 4620
Differential pinion	★	★		★		★		★		★		★		★			★	3120, 4620
Drive bevel ring gear	★	★	★	★		★		★		★		★		★			★	3120, 4620, 2315
Main drive bull gears	★	★	★	★		★		★		★		★		★			★	3145, 3120, 4620, 2345
Bull pinion	★	★	★	★		★		★		★		★		★			★	3120, 4620, 2515, 2315, 2345
Bull pinion shaft			★		★		★		★	★		★		★		★	★	3140, 3120, 2345
Sprocket shaft	★		★		★		★		★	★		★		★		★	★	3140, X-3140, 2345
Rear axle	★		★		★		★		★	★		★		★		★	★	3140, X-3140, 3250, 4340
Rear axle ring gears	★	☆	★	★		★		★		★		★		★			★	3145, 3120, 4620
Jackshaft	★		★		★		★		★	★		★		★		★	★	4620, 3140
Countershaft axles	★		★		★		★		★	★		★		★		★	★	3140, 4620
Steering knuckle pin		★	★	★	★			★		★		★	★		★			3120, 4620
Front axle vertical shaft	★				★		★		★	★		★		★			★	3140
Steering gears and sectors		★	★	★				★		★		★						3120
Motor studs and bolts	★										★		★	★	★			3135, 3140, 2330
Connecting rod bolts	★	★			★						★			★				3140, 2330
Body studs and bolts	★	★	★		★			★		★		★		★				3140, 2330
Roller bearings				★		★		★		★	★		★			★	★	3120, 4620

alignment also produces deflection in shafting which may result in badly overloaded bearings, if not shaft failures.

Having once obtained rigidity and alignment in proper shaft mounting, this condition must be maintained by the selection of a steel suitable for the bearing in which resistance to wear and shock as well as a high yield point in compression are essentials. These properties are met by S.A.E. 4620, an alloy commonly used for taper roller bearings. This alloy steel is resistant to wear and has a long true running life, thus contributing to the maintenance of proper shaft alignment. The core toughness of this material also provides the desired ability or capacity to resist shock.

Three factors have had a direct influence on gear design and materials for tractors. These factors are operating and service demands for:

1. Quieter operating units
2. Higher road speeds
3. Increased power

To accomplish the first, tractor engineers have specified that gear teeth be cut by improved methods. Also, whenever possible, the size of the gear has been reduced, which coupled with increase of power has led to the selection of stronger gear steels. Introduction of high pressure lubricants has made it possible to further increase tooth loads, which in turn introduces another factor demanding stronger steels.

Increased engine speeds have recently become an added consideration necessitating greater gear reduction which increases the problem of producing quiet power transmission.

Spur gears are used primarily because the permissible wide area of tooth contact makes possible the transmission of comparatively heavier loads. In meshing, the tooth action of a perfectly designed spur gear is that of rolling contact as distinguished from sliding contact of the hypoid type. However, because of volume change in heat-treatment, deflection under loading and possible errors in cutting, a slight departure from the ideal tooth contact may result in small areas of sliding contact which under light loads are harmless, but which under the high compressive loads encountered in tractor gearing might result in failure of the surface layers of the tooth from crushing, pitting, or spalling. To avoid this type of failure the tractor designer has chosen steels, usually carburizing types, capable of furnishing relatively high surface hardness.

Notch effect acting as a stress raiser is a factor to be considered particularly in the manufacturing operation. Tool marks which introduce localized areas of stress concentration in service can be the source of fatigue failure. Critical wear at the pitch line or area of highest pressure, concentration of stresses at the surface of the tooth under bending loads where the stresses exceed the proportional limit of the surface structure, and insufficient core hardness to adequately back the case under compressive loads are possible causes of failure in tractor gears. Concentration of loading caused by the shifting of contacting area from a full bearing across the tooth face to a small highly concentrated load on the end of the tooth is also a source of pitting fatigue failures. This last defect can be avoided by proper alignment of the shafting on which the gears are mounted and also by the selection of proper bearing size and materials.

Low crushing loads at point of tooth contact depends on many design factors. At best, designing with alloy steels may decrease the size of the gear to only a limited degree since there is an optimum stress for even the best alloys. Under commonly encountered conditions compressive stresses, as determined by the Herz method, range from 125,000 lb. per sq.in. on transmission gears to as high as 250,000 lb. per sq.in. on final drive gears in track-laying type tractors. Under certain operating conditions this maximum value has been reported to reach 300,000 lb. per sq.in., but only for short periods of operation.

In addition to the factors enumerated, shock resistance must also be considered for the ability of tractor gear steels to resist suddenly applied loads is of great importance. Shock loadings may be encountered in engineering or plowing service of a magnitude difficult to calculate except by empirical means.

Another factor which the tractor designer must take into account is the effect of operating temperatures on gears. Since a tractor is in service the year round, gears and other parts are subjected to temperatures ranging from 40 deg. F. below zero to temperatures reported to be as high as 400 deg. F. Sudden loads imposed on gears in starting a tractor that has been exposed to sub-freezing weather may result in breakage if suitable steels are not used. From service and laboratory tests it has been found that the presence of nickel in steel helps to retain most of the shock resisting properties when subjected to low temperatures.

Although temperatures up to 400 deg. F. have been reported as a result of the heat generated from the high com-

TABLE III—Tension Test Data

S. A. E. Spec.	Tensile Strength Lb. per Sq. In.	Yield Point Lb. per Sq. In.	Elongation Per Cent	Reduction of Area Per Cent	Surface Brinell
1040 W. Q....	96,000	61,000	24	49	186
3140 W. Q....	127,000	97,000	22	55	240
3240 O. Q....	147,000	118,000	19	47	290
4340 O. Q....	178,000	153,000	15	46	365

TABLE IV

Chemical Analysis of Alloy Steels Commonly Used in Tractors

S. A. E. Spec.	Carbon	Manganese	Nickel	Chromium	Molybdenum
3115.....	0.10–0.20	0.30–0.60	1.00–1.50	0.45–0.75	.....
3120.....	0.15–0.25	0.30–0.60	1.00–1.50	0.45–0.75	.....
3140.....	0.35–0.45	0.60–0.90	1.00–1.50	0.45–0.75	.....
X–3140.....	0.35–0.45	0.60–0.90	1.00–1.50	0.60–0.90	.....
3145.....	0.40–0.50	0.60–0.90	1.00–1.50	0.45–0.75	.....
3250.....	0.45–0.55	0.30–0.60	1.50–2.00	0.90–1.25	.....
4615.....	0.10–0.20	0.40–0.70	1.65–2.00	.....	0.20–0.30
4620.....	0.15–0.25	0.40–0.70	1.65–2.00	.....	0.20–0.30
4320.....	0.15–0.20	0.40–0.70	1.65–2.00	0.30–0.60	0.20–0.30
4340.....	0.35–0.45	0.50–0.80	1.50–2.00	0.50–0.80	0.30–0.40
2315.....	0.10–0.20	0.30–0.60	3.25–3.75	.....	.....
2320.....	0.15–0.25	0.30–0.60	3.25–3.75	.....	.....
2330.....	0.25–0.35	0.50–0.80	3.25–3.75	.....	.....
2345.....	0.40–0.50	0.60–0.90	3.25–3.75	.....	.....
4820.....	0.15–0.25	0.40–0.60	3.25–3.75	.....	0.20–0.30
2515.....	0.10–0.20	0.30–0.60	4.75–5.25	.....	.....

pressive loads coupled with atmospheric temperature, 250 to 300 deg. F. is more common. While the large size of the gears permits rapid conduction of heat away from the surface of the tooth thus providing a cooling influence, lubricants have heen known to become so thin from the temperatures developed that fuming was noted. Under such conditions the gear steels are put to a severe test.

Alloy steel gears are generally specified, except in a few instances where plain carbon steel will serve. The alloy carburizing grades are generally preferred to the medium carbon oil-hardening grades as greater wear resistance is obtained, as well as a higher endurance limit because of the high case strengths obtained in the carburized layer where stress concentration is greater under bending loads. Case breaking strengths range from 225,000 lb. per sq. in. in hardened S.A.E. 3120 gears to as high as 365,000 lb. per sq. in. in S.A.E. 4820 and 2515. Resistance to compressive loads is still higher. The degree of resistance to compressive loads depends to a certain extent on the type of heat-treatment. Direct quench from the carburizing temperature or a single quench to include both the case and core will produce a core structure sufficiently hard to form a suitable backing for the case. Either practice results in minimum distortion. Refining the case structure and leaving the core untreated should by all means be avoided as failure of the case is almost certain to result from crushing, particularly on such parts as pinions. Although the double quench treatment may furnish a slightly higher case strength, the core is softened considerably on reheating for the case treatment and little is gained in respect to resistance to crushing loads. The double treatment, however, provides slightly better wear as a result of the refinement obtained in the case, and does provide somewhat higher resistance to impact.

Properties Needed

Various degrees of strength, toughness and hardness are needed in a steel as a guarantee or insurance against the several factors contributing to failure of parts. Table V contains a list of the carburizing steels which have given satisfactory service in both the track-laying type tractor and the wheel type; core properties under various heat-treatments are also included.

Carburizing steels containing nickel have, in addition to their toughness and strength, a response to hardening which is of advantage in heat-treating. The presence of nickel depresses the critical hardening temperature range which in turn minimizes the tendency to scale or decarburize, also the distortion and volume changes in quenching and carburizing are held to a minimum. The latter being a very important factor in treating the larger gears.

A carbon range of 0.15–0.22 is considered desirable for alloy tractor gears. This range permits satisfactory machining of the blanks after controlled normalizing, while the carbon is sufficiently high to produce a uniform sorbitic core structure on quenching which is an advantage from the toughness and impact standpoint; also a deep case can be obtained. With carbon contents much over 0.22 the impact strength of the gear is lowered. With higher carbon cores the practice is to use thinner cases, which although permitting a saving in carburizing costs, has a lower resistance to crushing loads, particularly on the larger and more heavily loaded gears. To drop the carbon content below 0.15 is not usually desirable, since it is likely to result in a "stringy" machined finish and insufficient core hardness for proper backing of the case in the hardened gear.

For service involving rolling contact and extremely high unit loads it is generally accepted that a case of eutectoid composition or slightly higher at the surface is preferred. To obtain this carbon concentration a case depth of 0.045 to 0.055 in. is necessary, and can be readily obtained with the proper control. The presence of nickel also acts to hold excess carbides in the outer layer of the case to a minimum under the more economical carburizing conditions.

TABLE V—Physical Properties of Some Tractor Gear Steels

S. A. E. Spec.	Type of heat-treatment after carburizing	Core Properties: Tensile strength lb. per sq. in.	Yield point lb. per sq. in.	Per cent elong. in 2 in.	Per cent Red. in area	Hardness Brinell	Izod Impact ft.-lb.	Case hardness Rockwell C
3115, 3120	A	125,000	85,000	25	55	255	55	62–63
3115, 3120	B	155,000	125,000	13	43	335	30	60–62
4615, 4620	B	145,000	105,000	17	52	300	45	60–62
4615, 4620	C	135,000	105,000	20	55	290	65	59–61
2315, 2320	B	160,000	135,000	11	47	345	35	59–61
2315, 2320	A	95,000	60,000	35	65	195	85	60–62
4820	B	200,000	165,000	14	54	375	28	59–61
2515	B	195,000	160,000	14	50	385	30	58–60
2515	A	145,000	115,000	20	60	270	50	59–61

Heat-treatment: A — Cool from carburizing temperature; reheat to 1525–1550 deg. F., oil quench, reheat to 1400 deg. F., and oil quench.
B — Cool from carburizing temperature; reheat to 35–50 deg. F. above the core critical and oil quench. Treatment best adopted to fine grain steels.
C — Remove from carburizing box or gas carburizing furnace at carburizing temperature and quench in oil.

Treatment "C" is being increasingly used on 3115 and 4615 in tractor plants where gas carburizing is being employed, and is referred to as direct quench practice.

Single and double quench treatment continues to be used on 2315, 4820 and 2515 employed principally in crawler tractor construction.

Cleveland Tractor Company

ROAD SHOW DESIGNS of 1940

Details of New Construction Equipment Features

IMPROVEMENTS in fabrication methods, applications of new alloys, anti-friction bearings, synthetic seals and automatic controls are illustrated in the following design detail presentations of new road-building equipment exhibited at this year's Road Show.

Controlled welding is used extensively. Welded booms, built-up from formed sheets, are light in weight without any sacrifice in strength and therefore require less counterweighting at base. One manufacturer uses welded steel tubing for booms on shovels up to one cu. yd. in capacity, larger sizes have structural angle corner members and strut members of steel tubing. For complicated parts steel castings are generally used to facilitate production. The Thew shovel boom is combined weld-cast construction; sides are formed channel shape of varying depth, cover plates welded on and the cast steel X-member and cast steel end connection fork welded in.

Machine bases are welded or cast according to the individual designer's preference. Bay City machine bases are cast of nickel-manganese steel, heavily ribbed, since it is claimed that a good alloy casting will not only withstand torsion but will also absorb shock and vibration. On the other hand, the Harnischfeger machine base is built up around one deep-section formed sheet.

Forgings because of their uniform strength and machinability are used for highly-stressed parts such as sprockets, crawler shoes and table rollers. Crowd rack teeth are forged in short sections for easy replacement; bucket teeth are usually forged steel or cast manganese steel and are replaceable.

Flame-hardening and flame-cutting are used extensively. One trencher manufacturer flame-cuts the bucket wheel rim from one sheet of high-tensile steel, the inner circumference having a flame-cut bevel edge to provide a better seat for the forged teeth sections welded on later. This manufacturer flame-cuts buckets from the same sheet; they are later formed and welded. Sprocket wheels on one machine are of interesting construction: two flame-cut circles are dished and flame-cut for a cast steel hub which is welded in, bar stock is then formed and welded to the outer edges of the dished circles, and sprocket teeth are flame-cut through a template. Flame-cut teeth have a hardened surface.

To reduce frictional losses, Bay City and Link-Belt use grease-sealed ball bearings on all shafts and sheaves. Another shovel manufacturer Superfinishes main shafts. Pressure lubrication is through Alemite or Zerk fittings; Harnischfeger countersinks these fittings in the crawler assembly for protection.

Mechanical control with booster clutches is found in some equipment, others, such as LeTourneau, use preformed steel cable control, while Link-Belt and Harnischfeger have hydraulic control, the latter a newly-developed low-pressure system similar to automotive hydraulic braking.

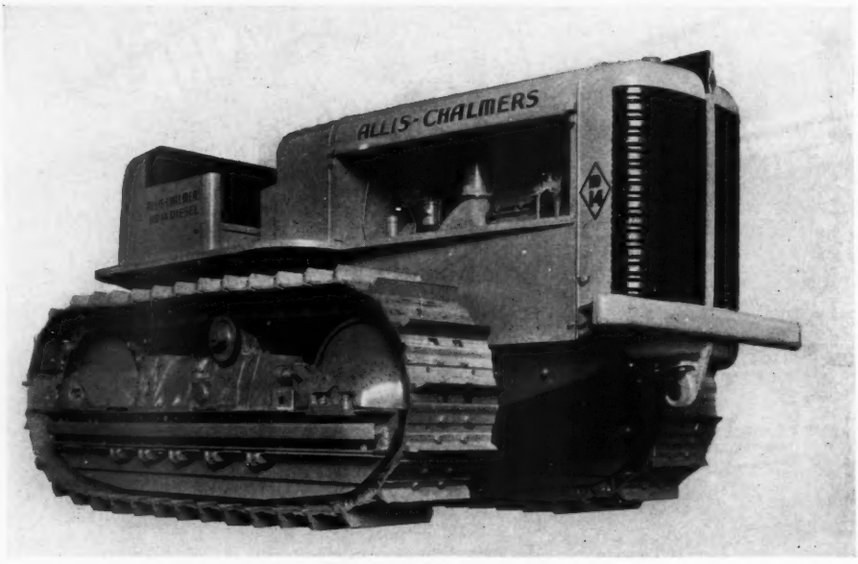
Power can be furnished by gasoline engines or diesels according to customer preference; the two-cycle General Motors diesel is finding wide acceptance because of its relatively small size and weight.

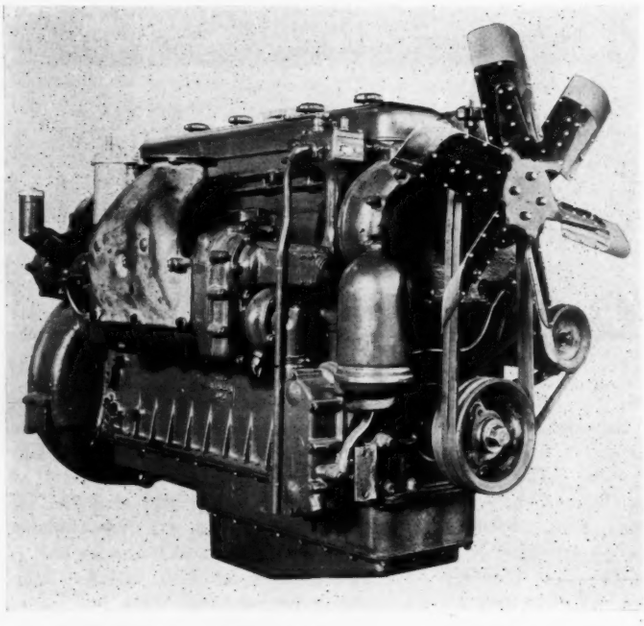
Space and time limitations naturally do not permit presentations of every product exhibited, but in later issues more outstanding construction machinery will be presented in our Modern Designs section. *(Continued on next page)*

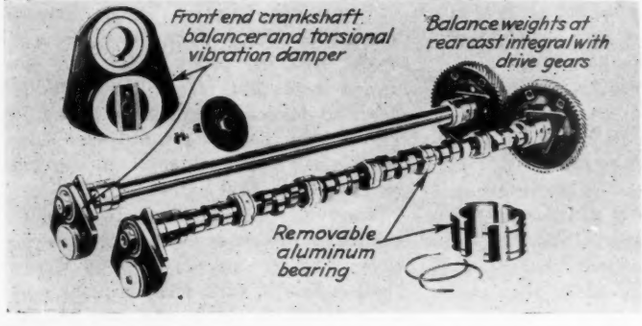
Two-Cycle Diesel, Graphite-Bronze Clutch Plates

POWERED by a General Motors six-cylinder, two-cycle diesel, this Allis-Chalmers HD 14 tractor weighs 27,900 lb. and exerts 108 hp. at the drawbar. Two smaller sizes in this series, having the same constructional features, are powered with four and three-cylinder diesels and furnish 71 and 54 drawbar hp. respectively. Transmission is of constant-mesh type. Spiral bevel gears are used.

Torque curve on the diesel is maximum between 800–1,200 r.p.m. Since normal operation is between 1,500–1,600 r.p.m., any overload slows engine down into range of maximum torque giving slightly higher drawbar pull. Variable-speed governor permits slowing engine to half speed without loss of drawbar pull, thus making possible use of throttle in any speed and eliminating considerable shifting. Shafts are Tocco-hardened.

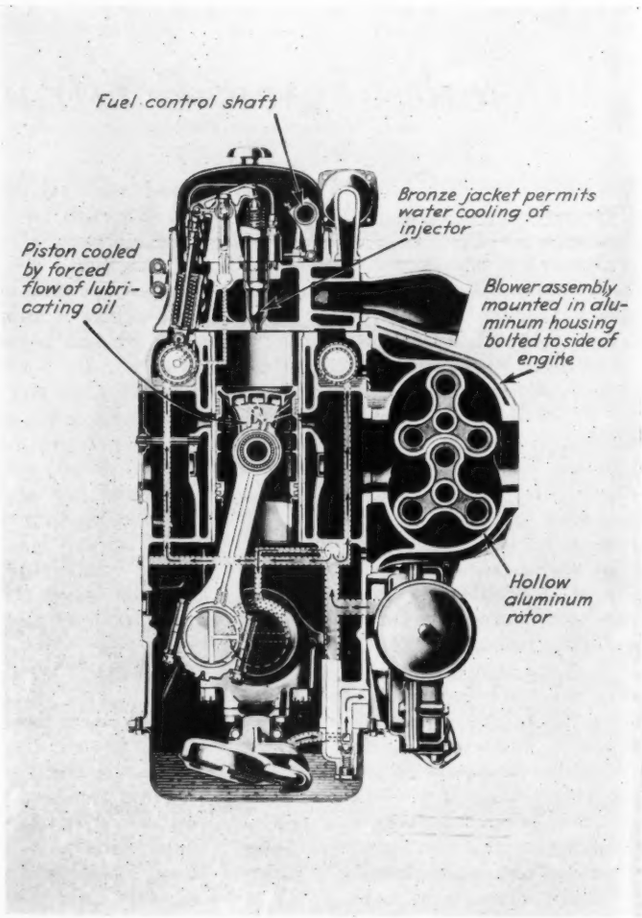


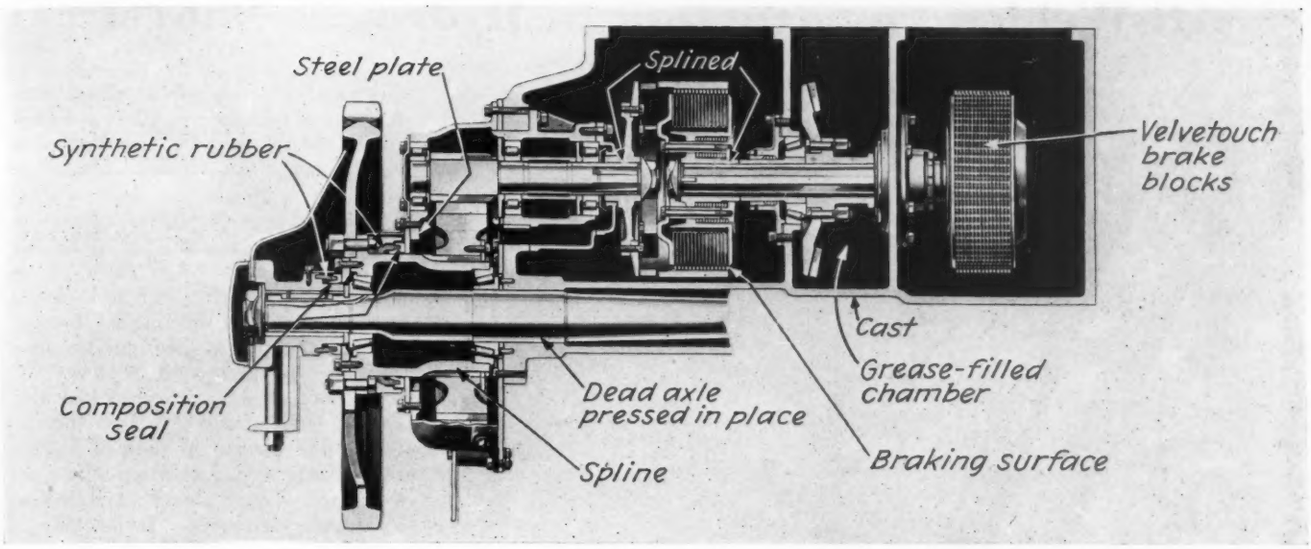



Front end crankshaft balancer and torsional vibration damper
Balance weights at rear cast integral with drive gears
Removable aluminum bearing

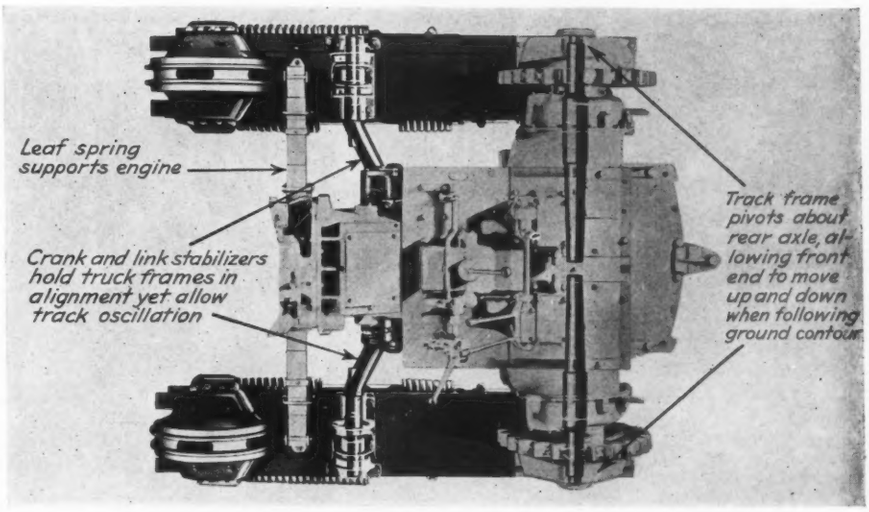
Balancing. Two weights, rotating in opposite directions with respect to each other, are used at each end of the engine. Rear weights are cast integral with camshaft and balance shaft drive gears. Front weights, articulated on hubs, are spring-loaded to damp torsional vibration. Replaceable aluminum bearings on camshaft are held in position by snap rings.

Pistons are cooled by lubricating oil supplied through the rifle-drilled connecting rod and sprayed against piston head. Two short grooves milled into connecting rod bearing shell are in constant registry with three oil-feed holes in the crankpin and supply a continuous stream of oil to a semi-circular groove in the connecting rod. A fixed orifice is used at end of rifle-drilled connecting rod. Piston pin is fitted with needle bearings.

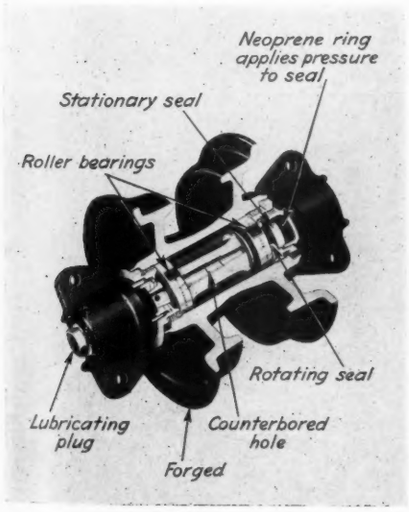

Fuel control shaft
Bronze jacket permits water cooling of injector
Piston cooled by forced flow of lubricating oil
Blower assembly mounted in aluminum housing bolted to side of engine
Hollow aluminum rotor

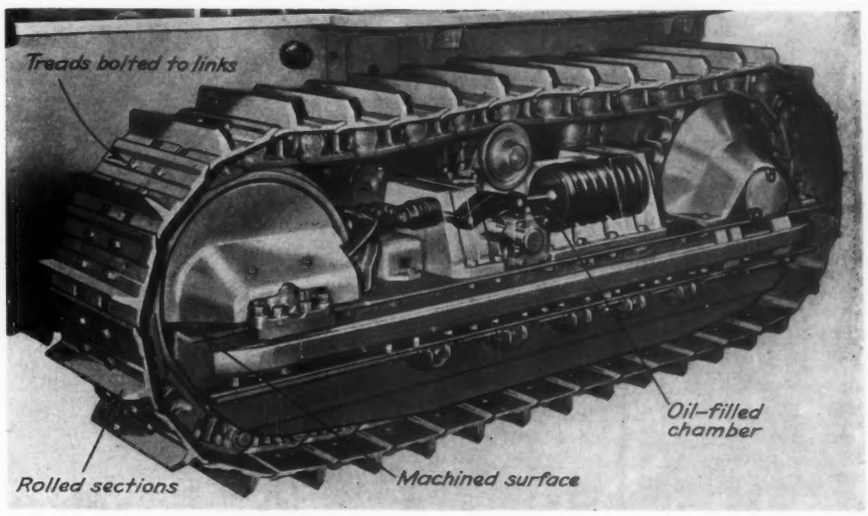


Steering clutch faces and steering brakes are lined with Velvetouch, a graphite-impregnated bronze material welded to a steel backing plate. Clutch driving plates are splined to shaft; driven plates are splined to clutch outer housing to which brakes are applied.


Leaf spring supports engine
Crank and link stabilizers hold truck frames in alignment yet allow track oscillation
Track frame pivots about rear axle, allowing front end to move up and down when following ground contour

Crawler treads pivot about rear axle and are supported against side sway by two crank and link stabilizers. Thus tracks can move up and down freely but always maintain alignment. Stabilizers carry no weight. This construction provides for necessary oscillation yet has strength of rigid track construction and even in large machines eliminates the need for the usual heavy tractor frame.





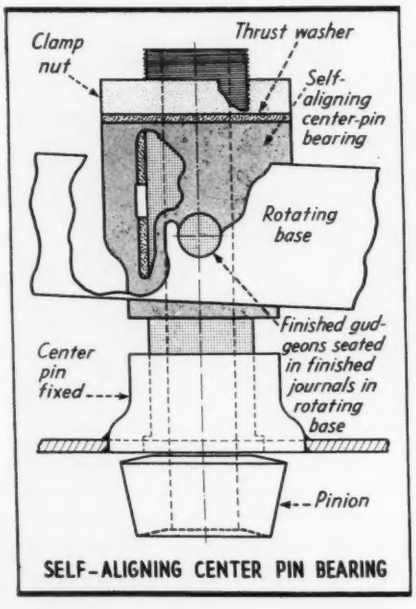
Positive-seal truck roller is fabricated of two forged parts welded at center and has bearings completely sealed by two steel rings held in tight contact by a ring of compressed neoprene. Lubricant is supplied through counterbored shaft.

Redesign of track release mechanism utilizes a bellcrank to apply spring pressure to the front idler. Front idler runs on roller bearings and has positive seals similar to those in truck rollers. Idler is a single steel casting having a large built-in oil reservoir. Track rails are drop forged and hardened to obtain additional wearing qualities and toughness. Track shoes have deep grousers. Sheet metal guards on underside of engine are standard on this tractor.

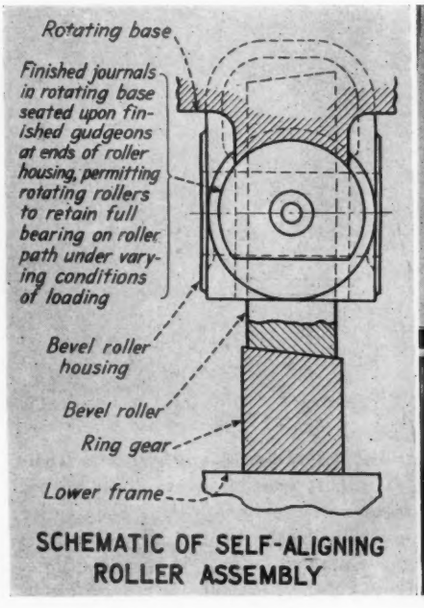
All-Welded Construction in Hydraulic Shovel

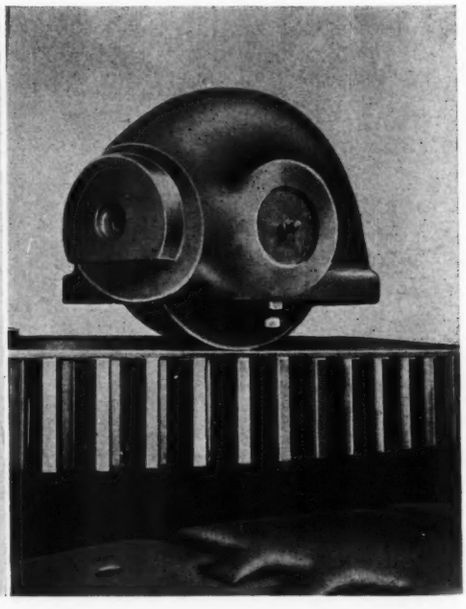
CONVERTIBLE to shovel, dragline, crane or trench-hoe service, the Link-Belt Speed-O-Matic has all principal parts welded and stress-relieved. Change of front end equipment is made without disturbing basic machinery. Change in speed for various types of front end equipment is made by replacing silent chain pinion on engine shaft or changing detachable split drum shells. Buckets are manganese steel. Bearings for drive axles are integral with crawler side frames and are bronze-bushed and dirt-sealed. Drive chain is taken up by sliding side frames on finished seat of cross members and inserting shims between face of cross member and face of side frame. Screw-type tension take-up is used on crawler.

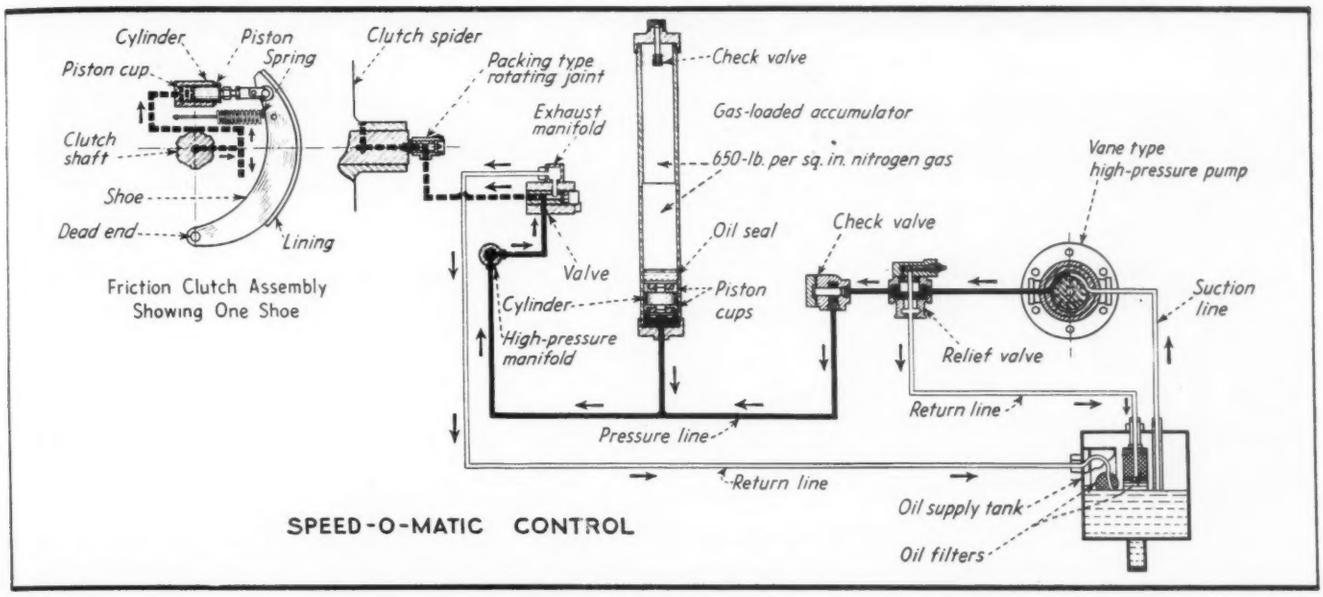
Self-aligning rollers and center-pin bearing. Forged and heat-treated steel rollers are housed in cages with trunnions which fit into the revolving table. This construction insures full face contact of rollers on roller path and eliminates concentrated loadings resulting from point contact as when conventional rollers are loaded eccentrically. Roller, roller path and bearing life is increased. Center pin of forged steel turns in two bronze bushings in a self-aligning center pin bearing in the rotating base. Center pin is fitted with anti-friction bearings, dirt-sealed. Vertical shaft is alloy steel heat-treated and splined. Two anti-friction radial and thrust bearings are used at top of vertical travel shaft, which runs inside of center shaft, instead of conventional thrust washer usually placed behind bevel gear pinion.

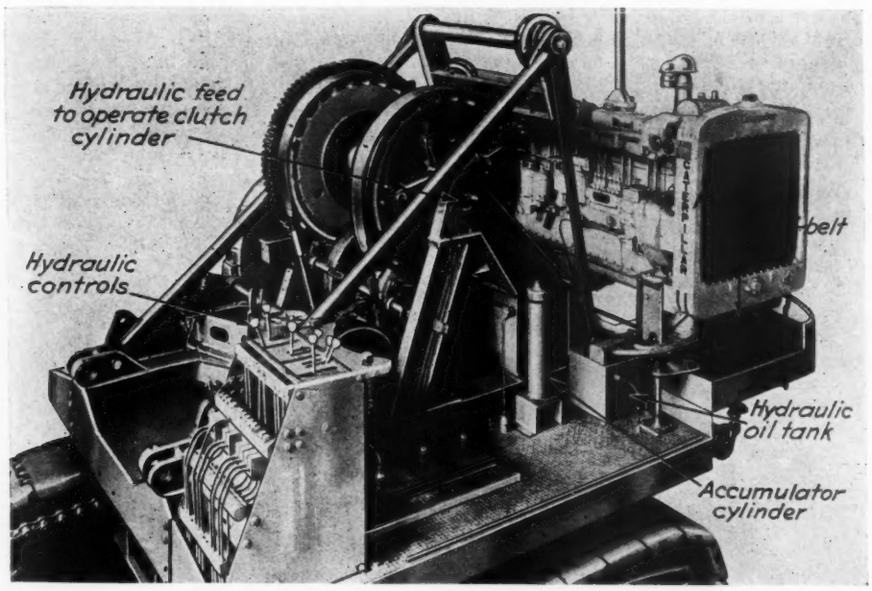


SELF-ALIGNING CENTER PIN BEARING

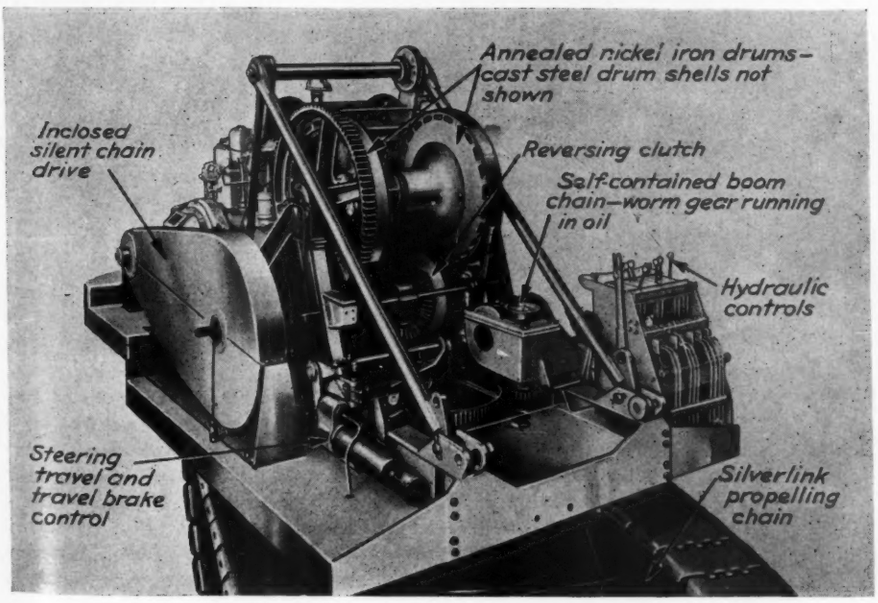

Rotating base
Finished journals in rotating base seated upon finished gudgeons at ends of roller housing, permitting rotating rollers to retain full bearing on roller path under varying conditions of loading
Bevel roller housing
Bevel roller
Ring gear
Lower frame
SCHEMATIC OF SELF-ALIGNING ROLLER ASSEMBLY







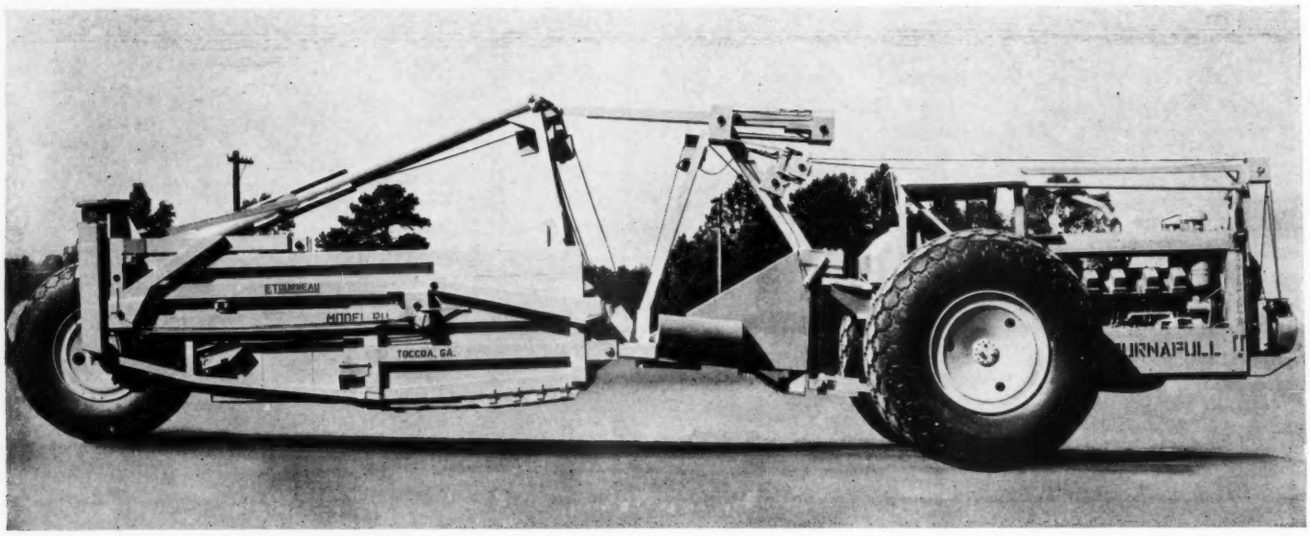
Hydraulic control is applied to all principal operations, including steering and traveling, through variable-pressure valves centralized on a single control panel directly in front of operator's cab seat. Pressure is supplied by a small hydraulic pump V-belt driven from the diesel or gasoline engine. Oil is delivered to a pressure manifold to which control valves are connected. A gas-loaded accumulator, oil-sealed, prevents formation of air pockets in the pressure line. Pressure oil connection to clutch and brake operating cylinders are through packing-type rotating joints. Brakes lock when machine stops.


Annealed nickel iron drums—cast steel drum shells not shown
Inclosed silent chain drive
Reversing clutch
Self-contained boom chain—worm gear running in oil
Hydraulic controls
Steering travel and travel brake control
Silverlink propelling chain

All-welded deep box-section lower frame is designed with a totally-inclosed oil-tight chamber for propelling gears and totally-inclosed chambers for steering and locking brakes and propelling clutches. This construction with integral cross members is twice the depth of the conventional design lower frame mounted on separate cross members. Entire frame is stress-relieved before machining. Drums are annealed nickel iron, bronze-bushed and mounted on heat-treated alloy steel shafts equipped with renewable cast-steel grooved shells.

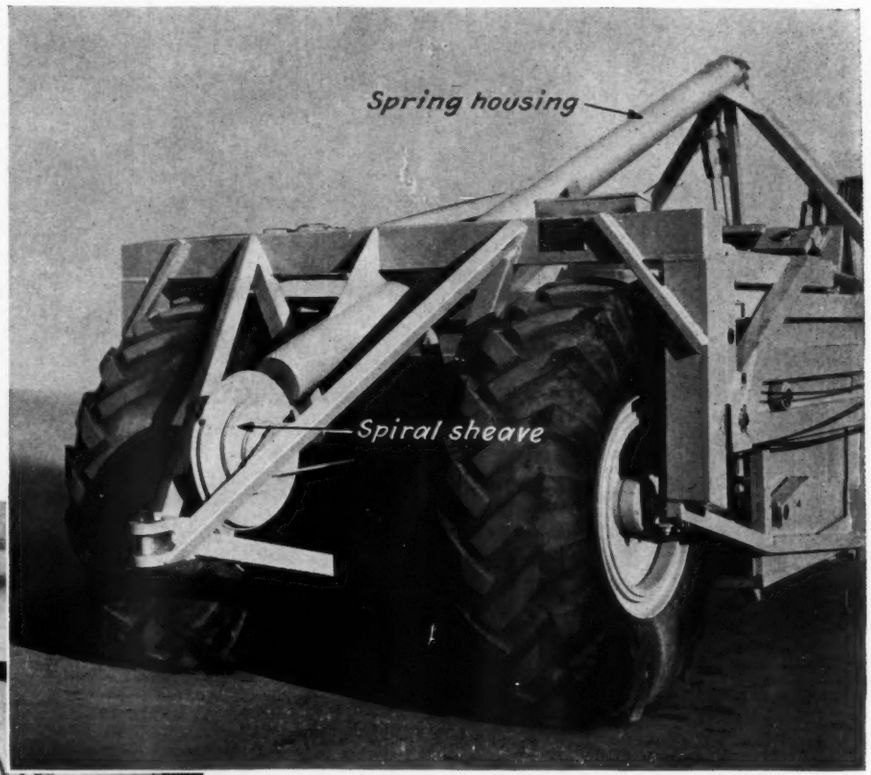
Propelling, steering and locking arrangement consists of a traction shaft, bevel-gear driven from vertical travel shaft, and fitted with roller chain drives to drive axles at one end of crawler side frames. Clutch and brake for each crawler are interconnected and operated by one lever. Slow-speed traction shaft runs in bronze-bushed bearings. Grease fittings on both lower and upper frames are grouped at a single point and marked to indicate kind of lubricant and frequency of lubrication required.

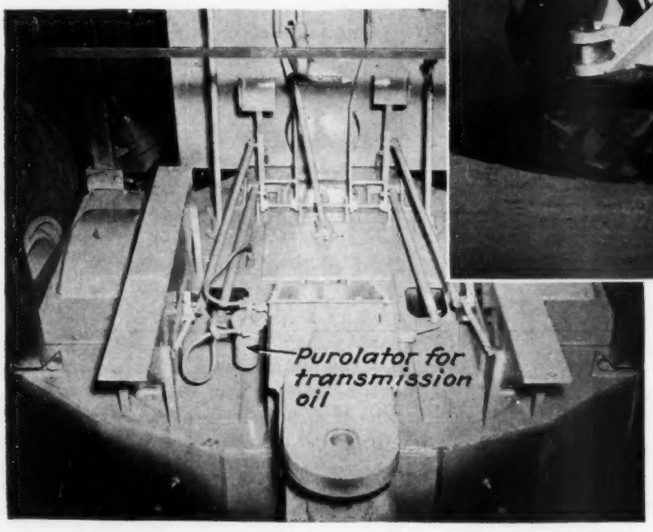
Diesel-Powered Scraper, Cable-Controlled



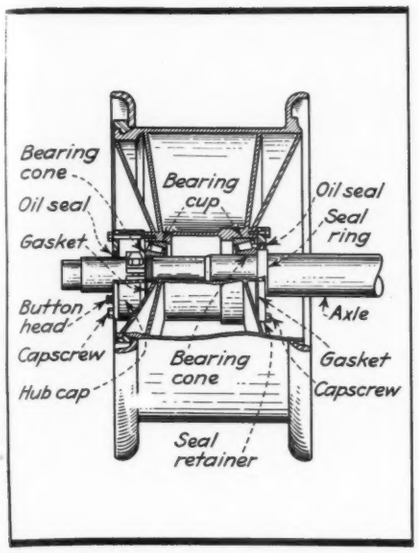
ROAD CONSTRUCTION COSTS can be reduced considerably if loaded materials are moved to the dumping ground as quickly as possible. The Le Tourneau Tournapull was especially developed to move materials rapidly, combining high loading capacities with truck speed for dumping. This Tournapull, one of the largest pieces of equipment shown at the Show, has a capacity of 45 cu.yd. and is powered by a 160 hp. Caterpillar diesel. Fully loaded, the machine can travel at 18 to 20 mi. per hr.; when empty, speed is 26 mi. per hr. Machine is steered by independent control of each wheel.

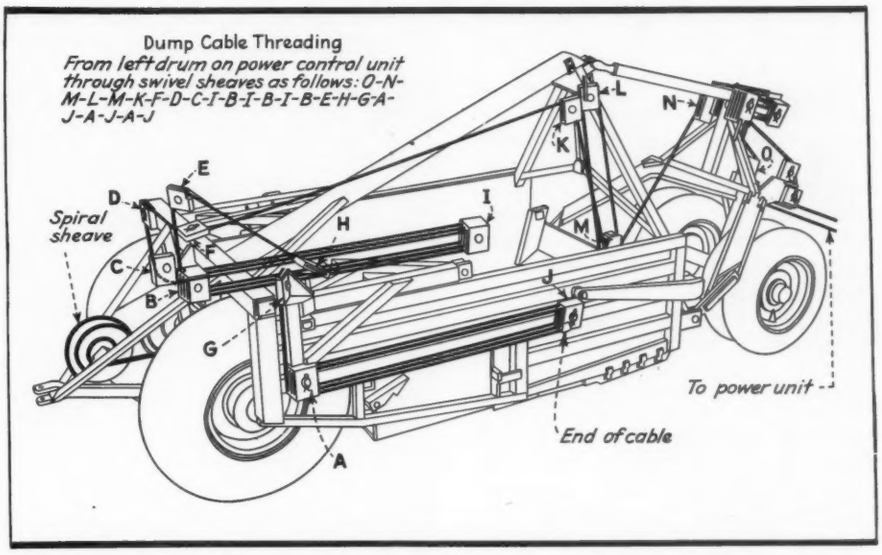
Rear view showing spiral sheave and coil spring housing. Scraper is arc-welded throughout. Sides of bucket and body are a special fairly high carbon content sheet steel, with good wearing and welding properties. Side reinforcing members are ship channel sections. Sheet and channels are welded in a special jig in which the plate is held in a curved position. When pieces are removed from jig and cooled they assume straight position. Bottoms of buckets on large carryalls are double thickness special alloy steel with a hardwood filler that acts as a buffer for rocks. Hard facing is used on the top of the reversible cutting blade.





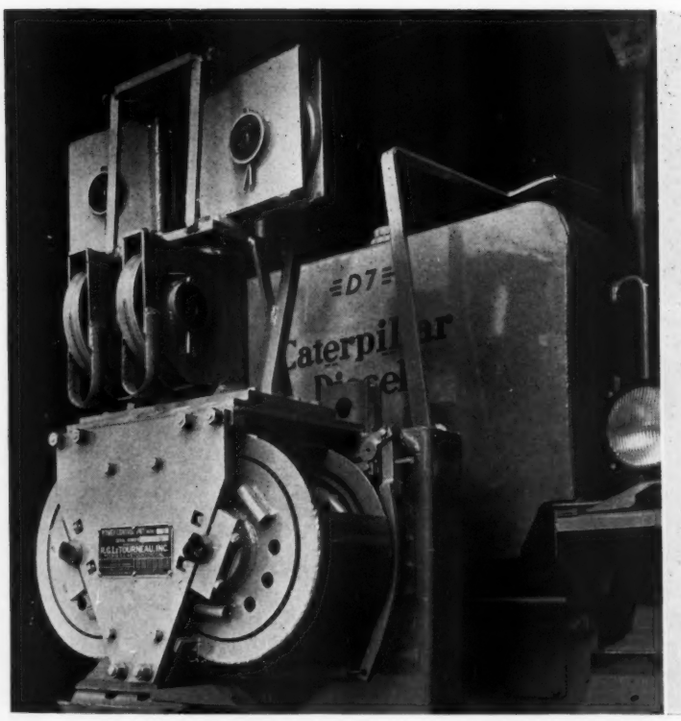
Transmission case for the driving gears and the wheel brakes is of welded construction and is stress-relieved. This case is on the aft end of the welded frame that supports the overhung diesel engine. Weight of engine plus most of load weight is carried on the front wheels equipped with 102 in. low-pressure rubber tires specially manufactured for this design. Wheels and gears are welded. Gear rims are rolled sheet, webs are circular-cut sheet, hubs are formed from sheet or bar stock.

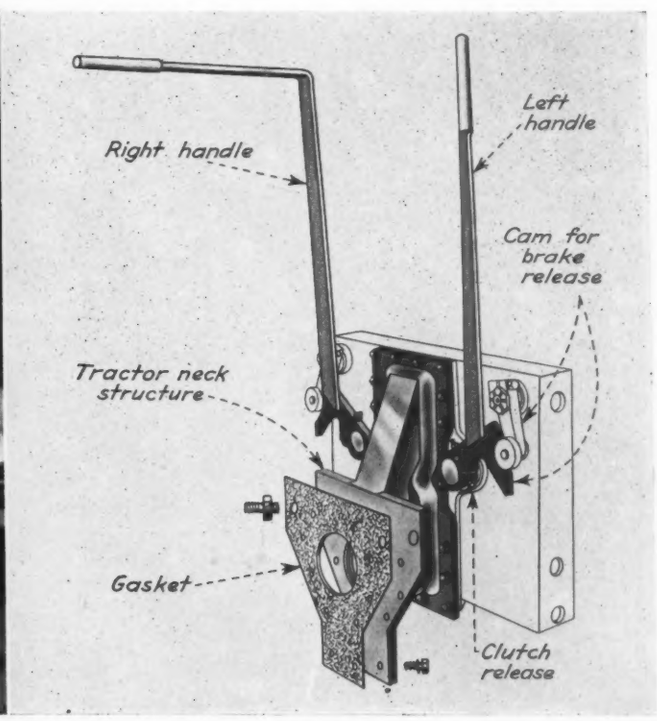



Dump Cable Threading
From left drum on power control unit through swivel sheaves as follows: O-N-M-L-M-K-F-D-C-I-B-I-B-I-B-E-H-G-A-J-A-J-A-J
Spiral sheave
To power unit
End of cable

Sheaves are fabricated of a center disk of alloy steel and two outer disks of mild steel, flame-cut from steel plate and welded together. Cable groove is machined in alloy steel center disk which is highly resistant to cable wear. Sheaves are mounted on roller bearings and have Zerk lubrication fittings.

Cable size and sheave size are standardized, sheaves being of 9½ in. tread diam. and cable being ½ in. pre-formed plow steel with independent wire-rope center. Standardization of cable size allowed standardization of power unit including gear reduction, brake drums and sheaves. The same power unit can be used on all equipment. Power cable pulls tail gate forward against tension supplied by a group of coil springs in a tubular housing. Cable running through the center of these springs to a pull plate passes over drum attached to spiral sheave, which maintains 2,000 lb. tension on tail gate at all times.





Power control unit, equipped with gear reducer, brake and clutch has two cable drums, the standard drum being 7⅝ in. in diameter and 7⅝ in. long, with a capacity of 186 ft. of ½-in. cable. Where more cable capacity is required, a special drum 14⅝ in. long is provided, having a capacity of 357 ft. of cable. All drums are welded and have 13-inch-diameter flanges.

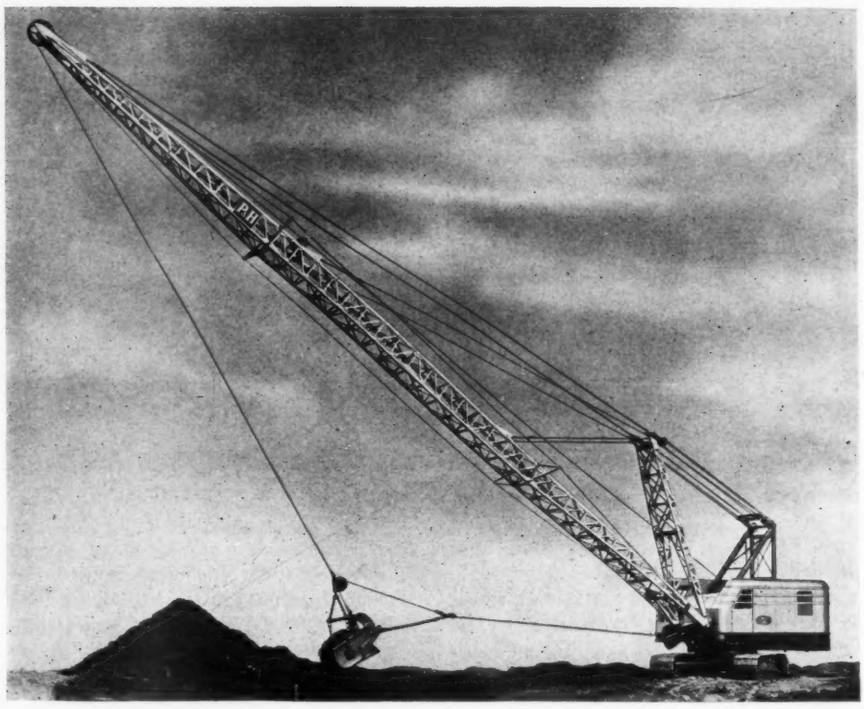
Each cable feeds from its drum to a spooling sheave, then to a fair-lead sheave, and thence to the scraper sheaves. The housing of each sheave is pivoted so that the sheaves can swing, allowing the cable to spool evenly onto the drum with a short lead, and the lead sheave and cable to remain in line with the first sheave on the scraper.

By locating the pivots of both sheaves directly in line with the vertical length of cable that feeds from one to the other sheave, the turning of either or both does not change the position of the cable between the two sheaves, and therefore there is no binding of the cable against the sides of the sheave grooves.

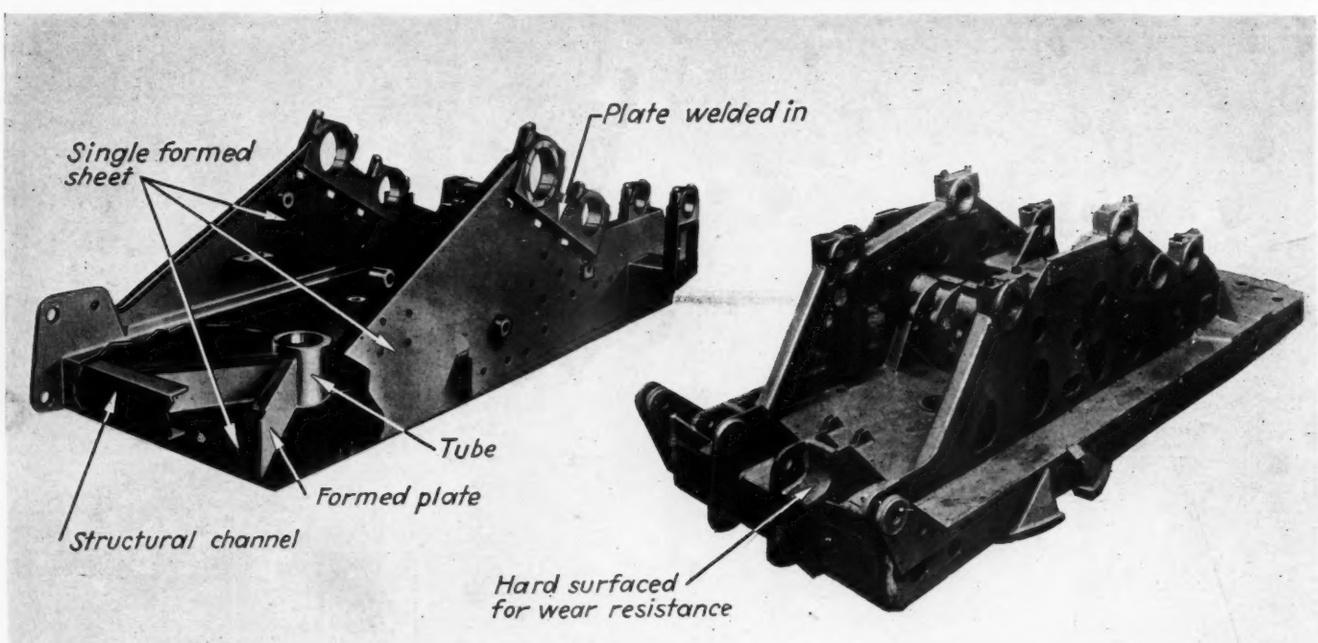
Extra length of cable is carried on a storage drum. As cable wears, worn part at power end can be cut off.

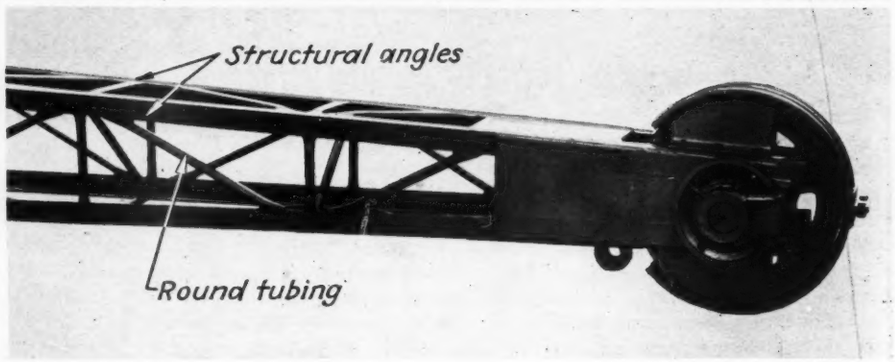
Low Pressure Hydraulic System Controls Shovel

HARNISCHFEGER SHOVELS, drag-lines, and other excavating equipment are now equipped with a low-pressure hydraulic control system which permits the operator to "feel" the action of the individual movements giving him accurate control. With the exception of drive sprockets on the crawler and a few intricate parts, the entire machine is fabricated of welded plate and structural steel. The shop has a press capable of forming 3 in. plate.

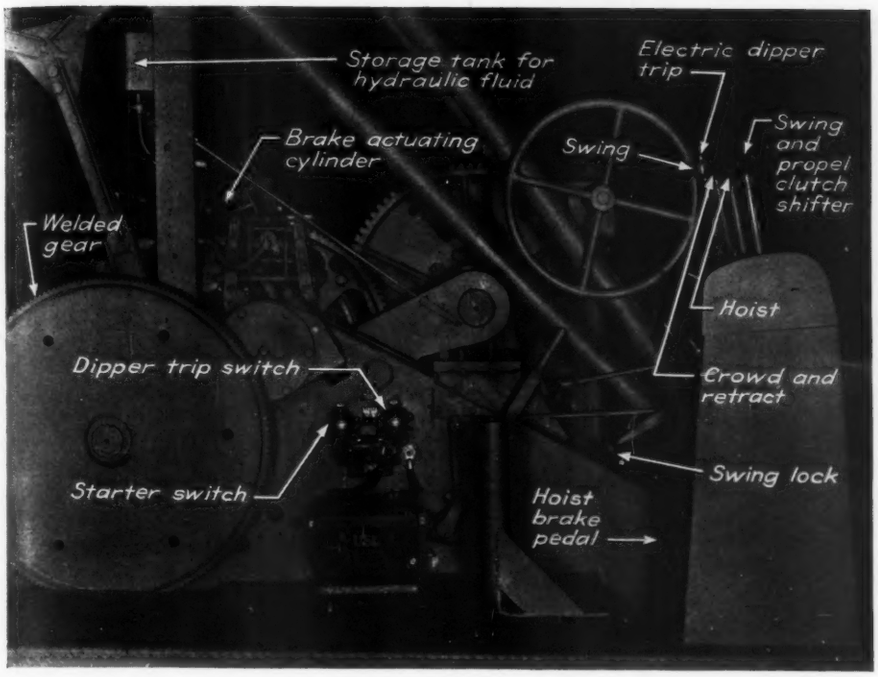


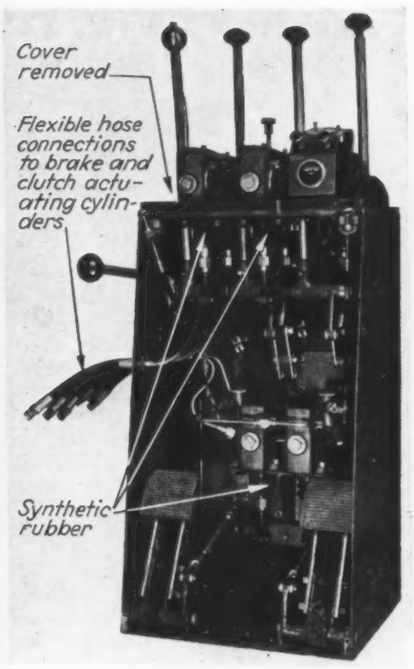
Alloy steels are used throughout. Low manganese steels are specified for abrasion resistance while higher manganese steels such as Manten are used for their strength plus low carbon content for weldability. Swing gear is welded and flame hardened. Flame hardening is used where the wear factor requires more than deep heat-treatment will provide. Usually 500 to 600 Brinell is desired. In some cases, gears are heat-treated for toughness and strength, and then are flame hardened for the necessary hard surface for wear resistance.



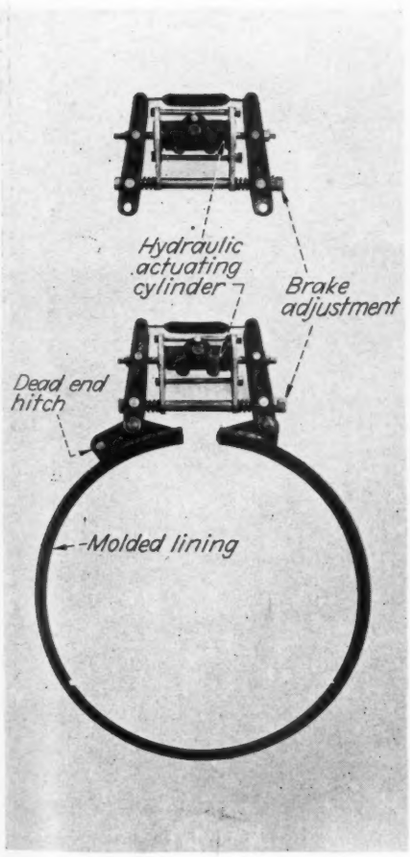


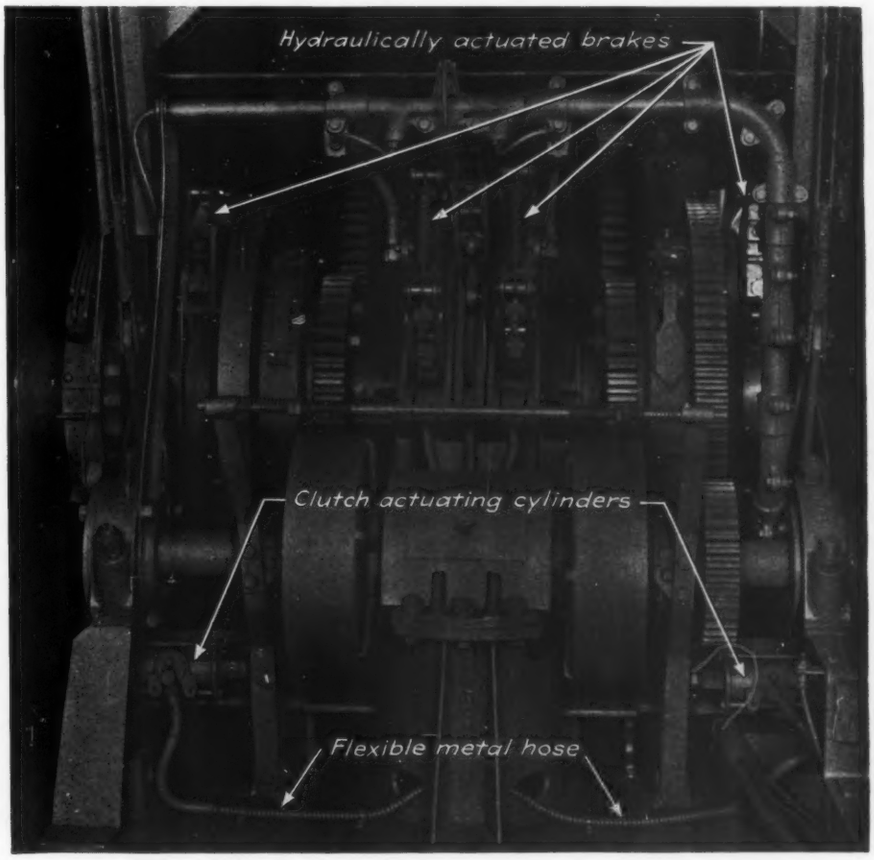
Steel tubing. For shovels of less than 1 cu. yd. capacity the boom is made entirely of welded tubing; on the larger shovels the booms are welded of structural angle corner-members and tubing for the strut members. Boom combines strength with lightness in weight, requires less balancing weight at base.





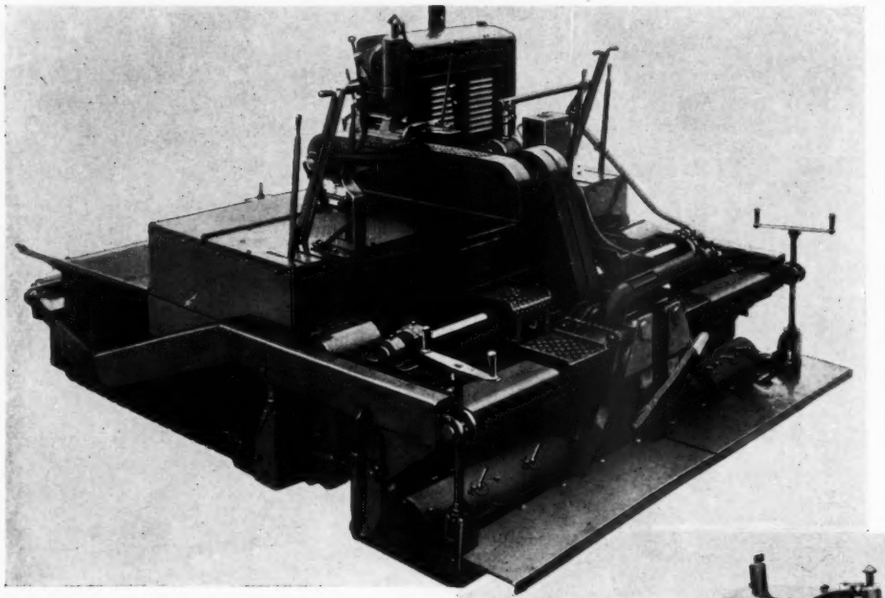
Hydraulic control is in effect a low pressure system with the operator acting as the prime mover as in heavy automotive truck braking hook-ups. There is an independent line for each motion. An accumulator set above the highest cylinder in the system keeps the lines full of oil at all times and no air can accumulate. No stuffing boxes are used. A return spring in the working cylinders is just strong enough to return the cylinder. Synthetic rubber boots protect piston rods.





Full-floating brake has only one dead end hitch. Method of applying brake-power makes shoe adapt itself to the contour of braking surface. There is only one nut for wear take-up adjustment. Clutches are engaged by a collar expanded by a sloping wedge cam moved by cylinders connected to control panel by flexible metallic hose.

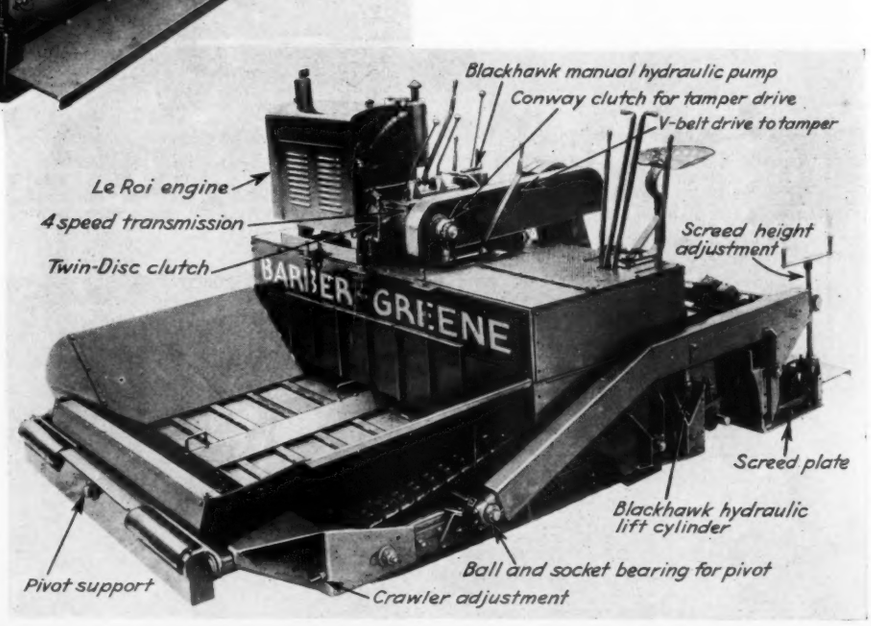
Finisher Lays, Tamps and Levels Bituminous Materials

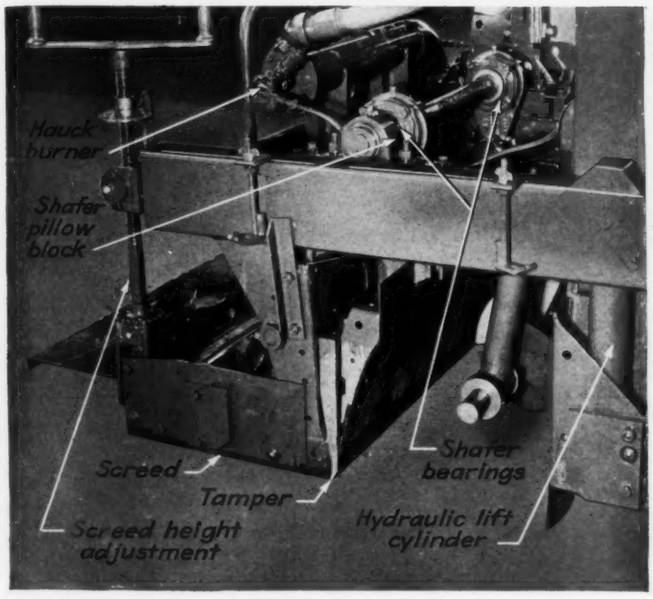


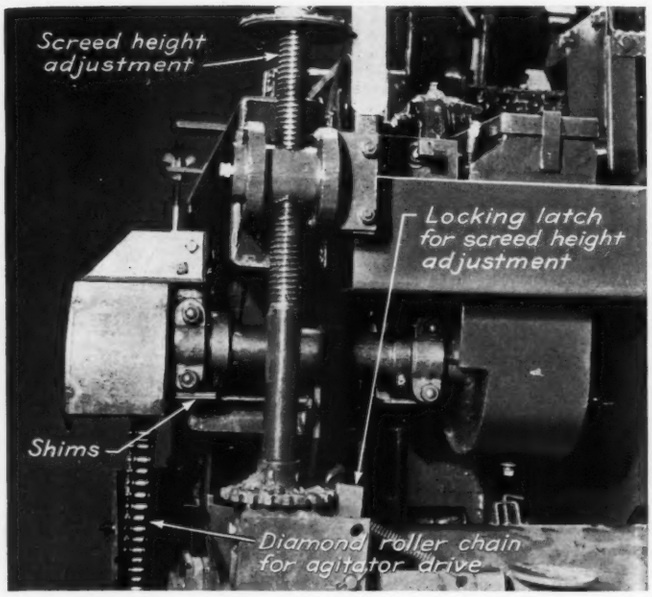
FITTED WITH DUAL CONTROLS for operation from either side, and driven by a 33 hp. LeRoi 4-cylinder gasoline engine through a 10 in. Twin-Disc clutch and 4-speed transmission, this Barber-Greene paving machine lays, tamps and levels bituminous materials to any desired thickness and to any desired crown without forms.

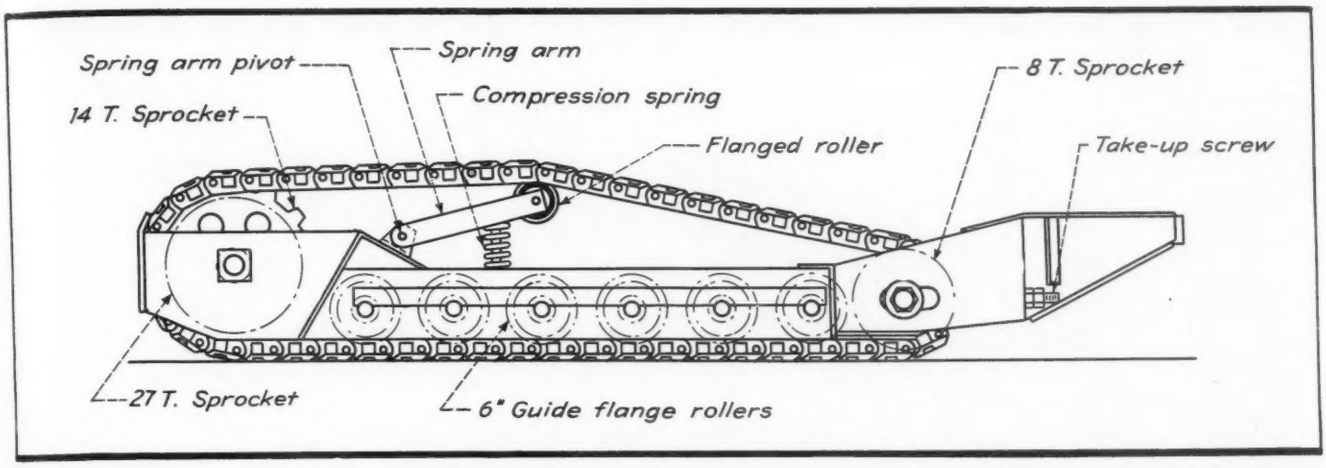
To allow for minor misalignments which occur in built-up constructions of this type, Barber-Greene-designed ball-and-socket bearings are used for shafts subject to misalignment of two or three degrees. For bearing pressures below 500 lb. per sq. in., these bearings are babbitted; for pressures above 500 lb. per sq. in., they are bronze lined.

Tamper is double V-belt driven. Tamper connecting rods are on an eccentric which give tamper $\frac{1}{8}$ in. reciprocating motion at shaft speed of 1,200 r.p.m. Tamper and screed are hinged at center to allow for crowning the road, hence V-belt drive is used as it has been found that minor misalignments do not appreciably affect drive efficiency. Tamper, shown below, left, is made of S. A. E. 52100 hardened steel. Mechanism is supported on two welded box beams which are lifted into traveling position by hydraulic jacks supplied with pressure oil from a manually operated Blackhawk pump unit. Screed, below right which acts as an ironer, is made of Manten steel. Sprocket for Diamond chain drive is case-hardened mild steel.



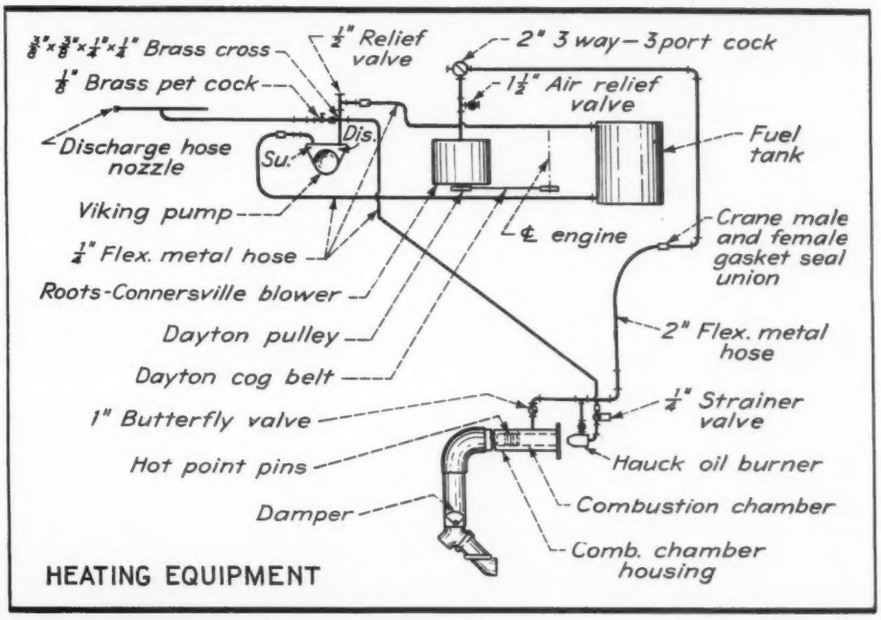




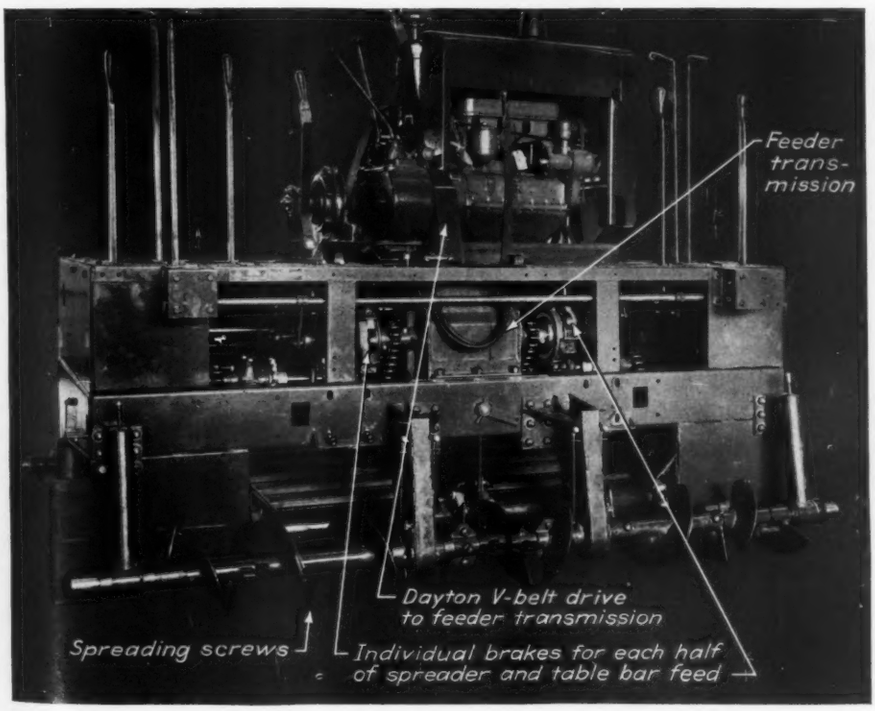


Tension in crawler is maintained by a roller held against links by a compression spring. This prevents breaking of treads if dirt or foreign materials jam between sprockets and crawler links. Take-up bolts are located at front of crawler frame. Drive chains are taken up by shims placed beneath drive shaft bearing. Crawlers are supported at front end by a box member pivoted at center to main frame.

Controlled hot air is used to warm screed when handling tacky materials. Heating circuit, shown schematically, consists of a blower and fuel pump V-belt driven from the front end of engine. The screed itself acts as the lower wall of the heating chamber. Air and oil are supplied to the burner through flexible tubing. Baffles in combustion and heating chambers permit even distribution of heat over full area of screed. Burner consumes standard fuel oils.



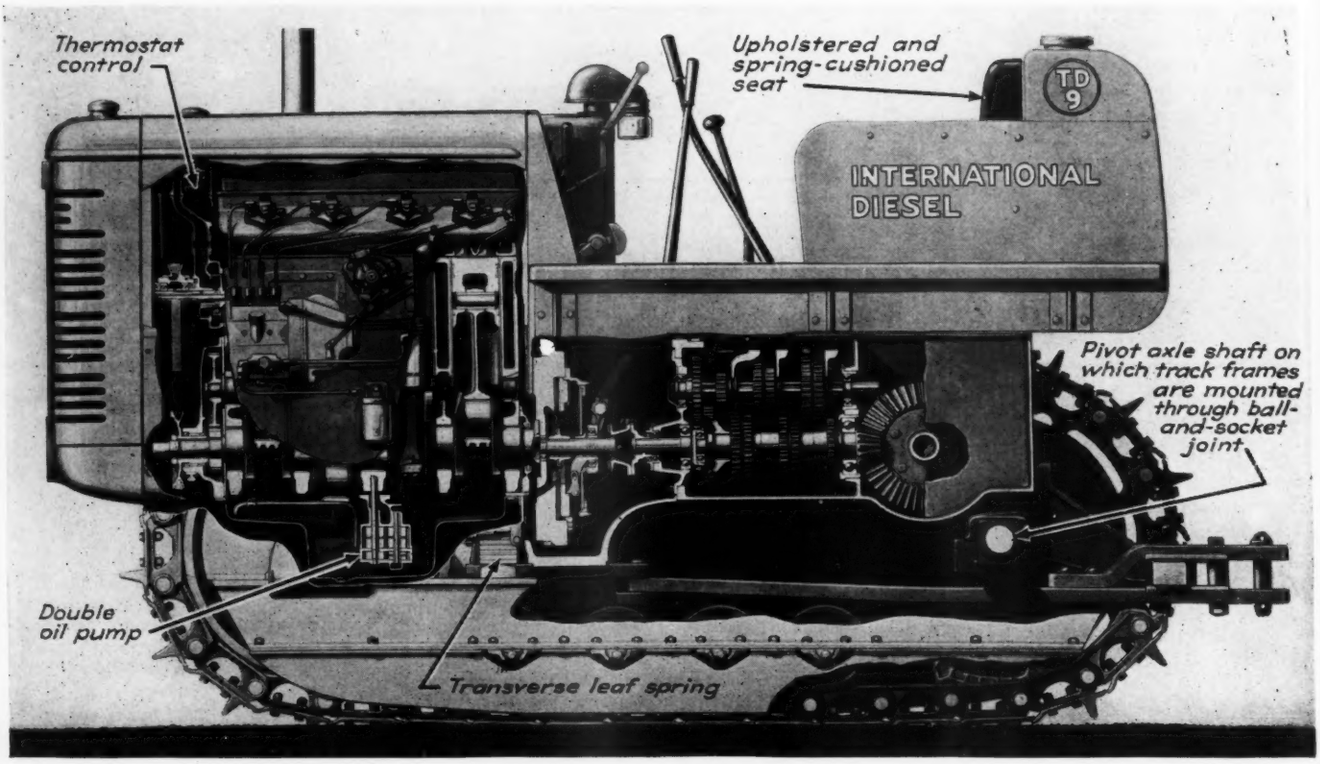
HEATING EQUIPMENT


Feeder transmission
Dayton V-belt drive to feeder transmission
Spreading screws
Individual brakes for each half of spreader and table bar feed

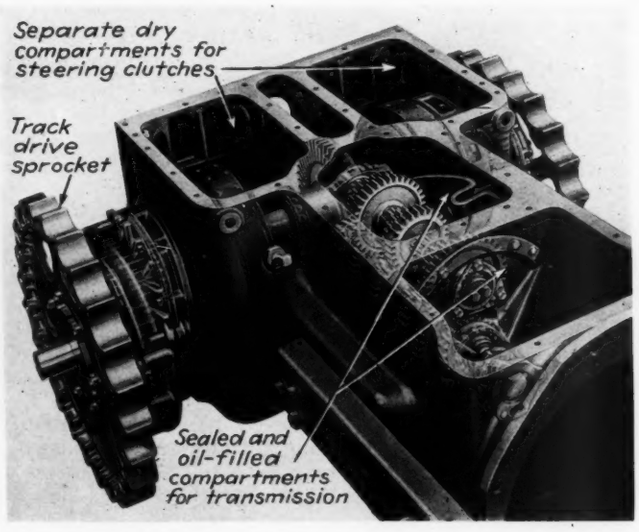
Material spreading screws, of welded construction, are 0.40 to 0.50 carbon steel with a Stellite edge. Screw is roller-chain driven at 18½ r.p.m. from main gearbox. Screws are mounted on two independent shafts, in line. Right hand and left hand thread screw construction moves the material from the center out to full width of machine. Worn spirals and paddles can be replaced easily.

A NEW, LONG-LIFE FRICTION LINING for brakes and clutches is now standard equipment on Allis-Chalmers tractors and optional equipment on LeTourneau scrapers. Consisting of a graphite-impregnated bronze material welded to a steel backing plate, this new lining, while expensive, has been adopted because its long life greatly reduces maintenance costs.

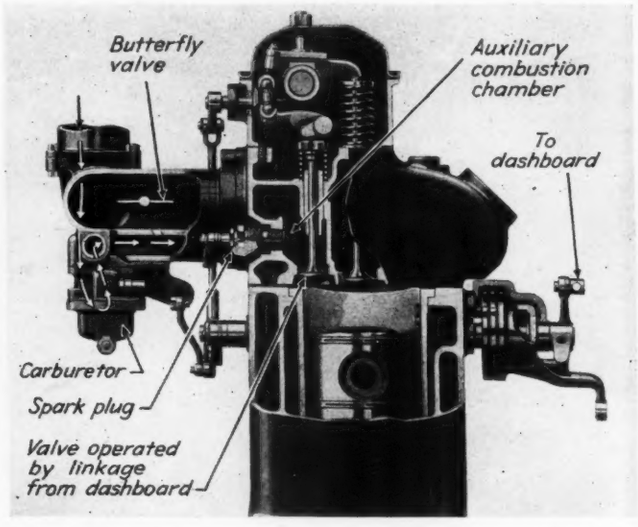
Steel Channels Support Engine on Cast Iron Frame


Thermostat control
Upholstered and spring-cushioned seat
TD 9
INTERNATIONAL DIESEL
Pivot axle shaft on which track frames are mounted through ball-and-socket joint
Double oil pump
Transverse leaf spring

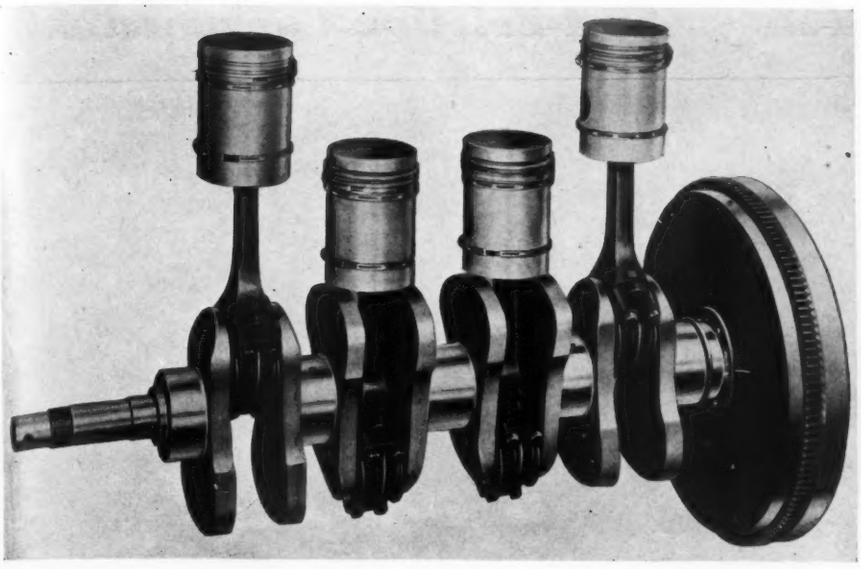
FIVE FORWARD SPEEDS are available in the new models *TD*-6, *TD*-9 International Tractractors. High and low reverse speed reduce time losses in backing. *TD*-14 has six forward speeds. All working parts are inclosed in dust-proof and water-tight housings. The tractor is powered by an International diesel equipped with an auxiliary combustion system by means of which the diesel is started and warmed up as a gasoline engine. A double oil pump and full pressure system assure positive lubrication of working parts regardless of grade on which tractor may be operating. Oil under pressure is conveyed through rifle-drilled passages in the crankshaft, connecting rods and engine block to bearing surfaces. Cleaners are supplied for air, fuel and lubricating oil. Centrifugal water pump, thermostatically controlled cooling water circulation and direct cooling of valve seats are additional features of the diesels powering these tractors.


Separate dry compartments for steering clutches
Track drive sprocket
Sealed and oil-filled compartments for transmission

Heavy one-piece cast iron frame houses transmission, bevel gears, steering clutches and steering brakes. Two heavy rolled steel channels, doweled and bolted to the front of the main frame, support engine, thus engine is not subjected to frame stresses. Main frame casting has dry compartments for steering clutches and brakes. Clutches are accessible through cover plates; brakes are adjustable externally.

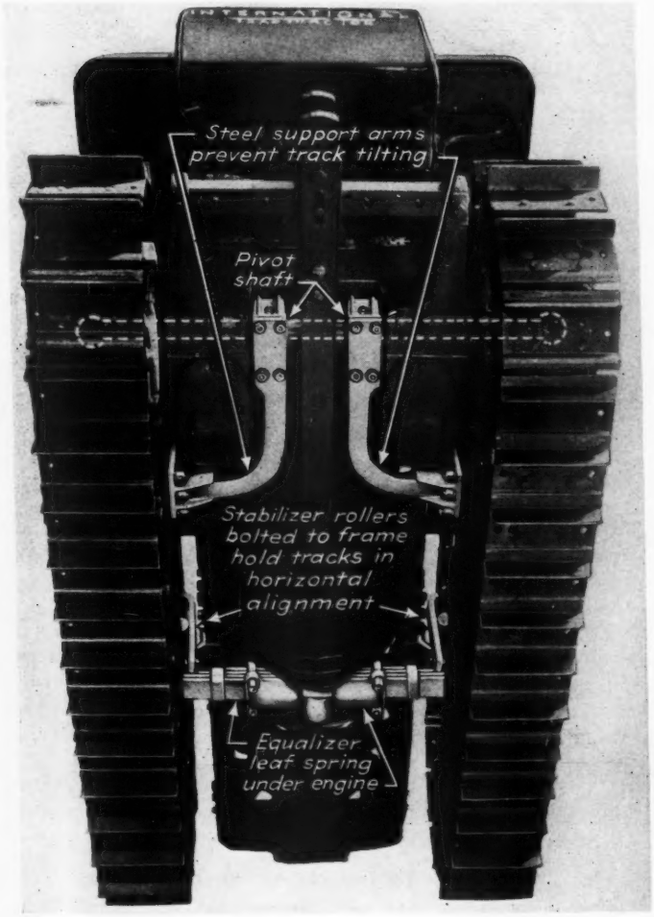


Starting of diesel as a gasoline engine is accomplished with an auxiliary chamber fitted with a spark plug and connected to each cylinder. Operation of dashboard lever opens valve to auxiliary combustion chamber, reducing compression, and at the same time closes the butterfly valve shunting air supply through the carburetor used for gasoline starting. After warm-up, lever is returned and engine is operated as a full diesel.

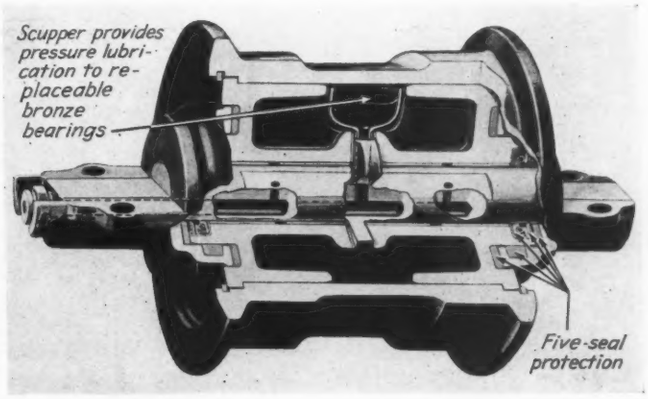


Five bearing crankshaft is heat-treated with main and connecting rod bearing surfaces Tocco-hardened. Main and connecting rod bearings are replaceable. Aluminum-alloy pistons are fitted with four compression rings and an upper and lower oil ring. Full-floating, pressure-lubricated piston pins are carburized, hardened, ground. Crankshaft is dynamically and statically balanced.

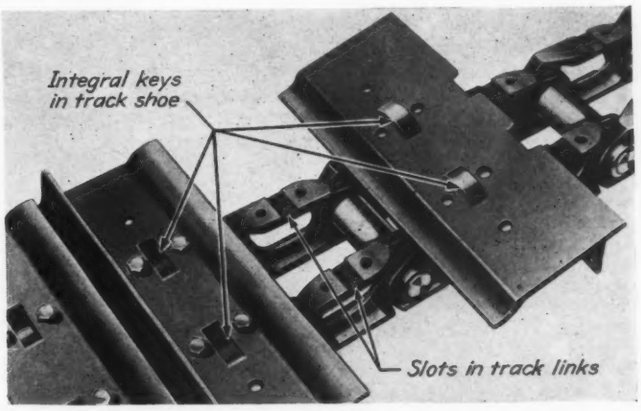
Cylinder sleeves are individually cast of electric furnace iron, heat-treated to 500 Brinell, and are honed and polished on inner surface. Matched sleeves and pistons are available for replacement.


Steel support arms prevent track tilting
Pivot shaft
Stabilizer rollers bolted to frame hold tracks in horizontal alignment
Equalizer leaf spring under engine

Track frame stabilizer maintains positive track alignment and relieves pivot shaft and track frames of torsional stresses. Frames are mounted on pivot shaft with ball and socket joints. Stabilizer rollers hold tracks in lengthwise alignment; steel arms mounted on pivot shaft maintain vertical alignment.



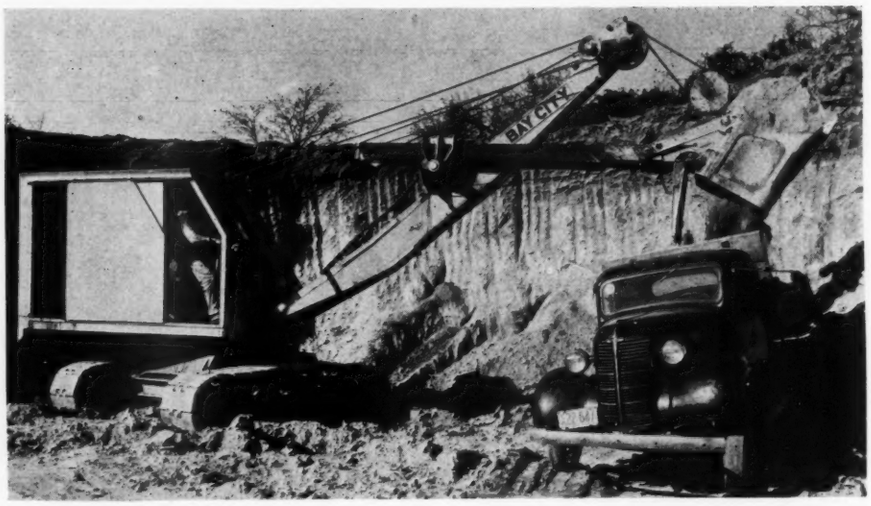
All track rotating parts including sprockets, idlers, and upper and lower track rollers are equipped with multiple seals. Replaceable bronze bearings are lubricated by oil scupper.



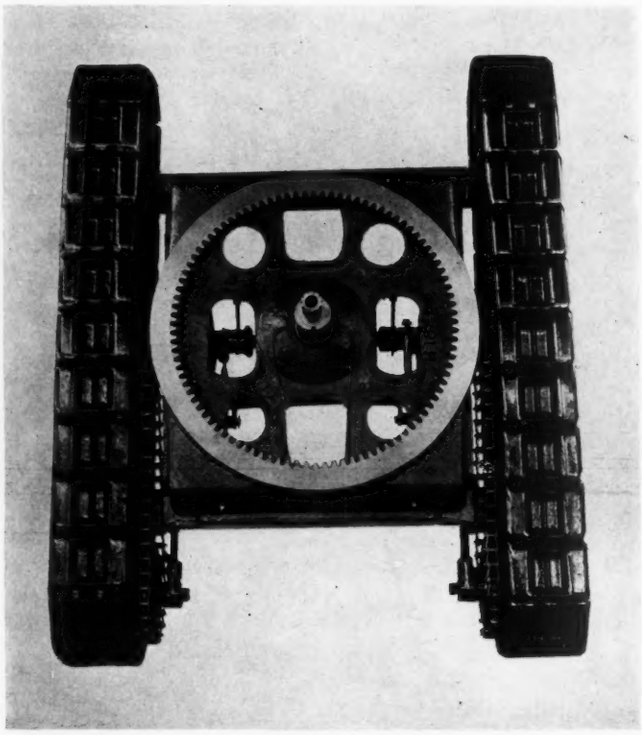
Keyed and bolted track shoes. Formed keys integral in the track shoes fit firmly into slots in the track links. Bolts hold shoes securely against links. Twisting and shearing action encountered in service is transmitted through the keys relieving stress on bolts and also preventing loosening of shoes.

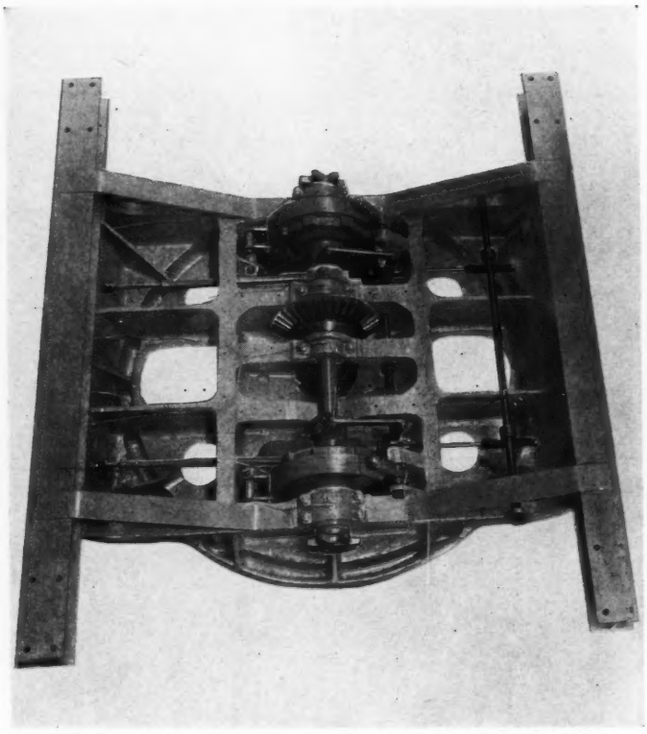
Alloy Castings, Forgings, Anti-Friction Construction

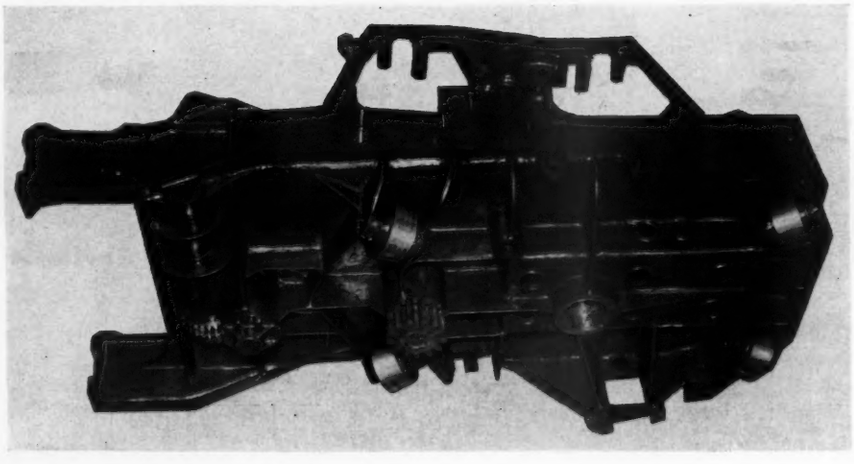
ANTI-FRICTION bearings throughout, heat-treated and normalized alloy steel castings and extensive use of drop-forged parts are among the design features to be found in the new Bay City shovels. Moving parts are pressure-lubricated through countersunk fittings.



Base castings of nickel-manganese steel are normalized, then totally heat-treated by a liquid quench and draw. These heavy alloy-steel castings absorb shock and vibration and are ribbed and cross-braced to resist twisting and distortion. Four large-diameter conical rollers, drop-forged, support revolving table. Splined propelling shaft operates inside centerpost cast integral with lower body.

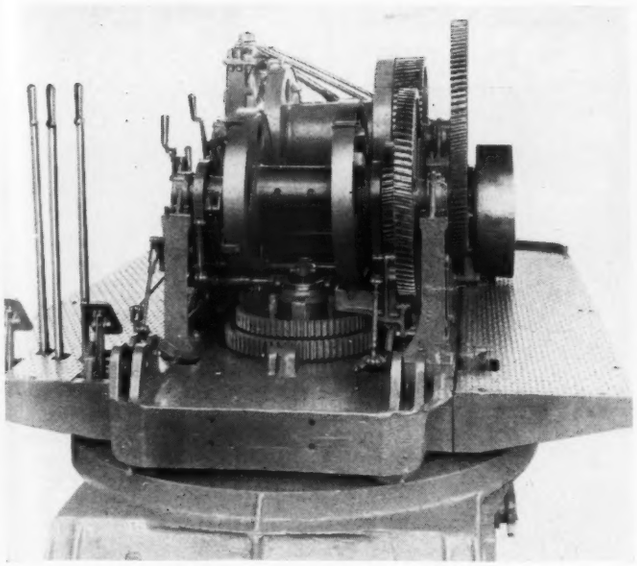




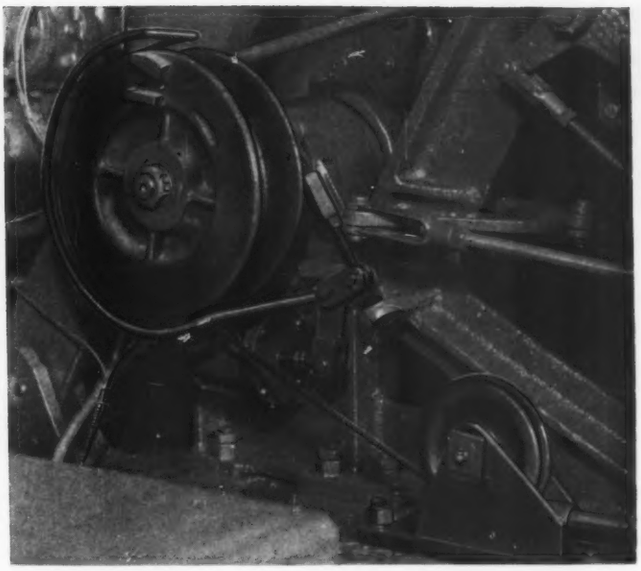


Steering with machine in full motion is accomplished with booster clutch control of crawler brakes. Operator need only set small cone clutch; power required to operate brake is transmitted from revolving shaft through booster clutch to brake arm. This eliminates stopping to shift jaw clutches and set hand brakes.

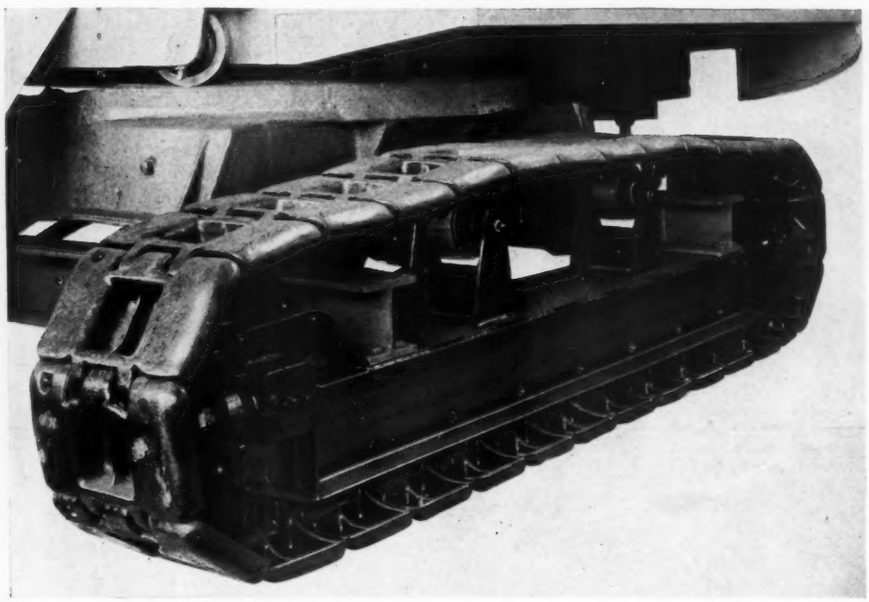
Underside of unit-cast alloy-steel revolving table on which all machinery is mounted, showing safety worm and worm wheel boom hoist, hardened alloy steel swing pinion splined to vertical shaft and bronze bushing for center pin. Side frames for machinery shafts are cast integral. Engine is mounted on this casting.

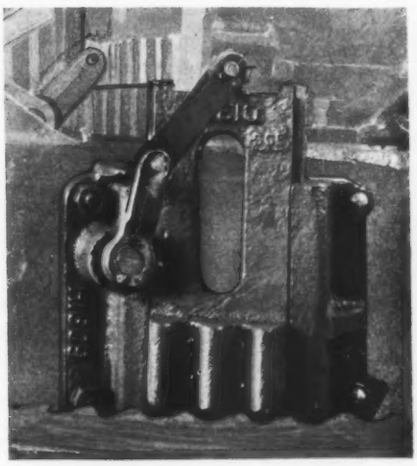


Helical-cut gears run in oil and are splined to the S.A.E. 4150 alloy shafts. Shafts and boom point sheaves are fitted with grease-sealed ball bearings and dirt seals. Boom foot lugs cast integral with base are directly over conical rollers.



Electric dipper trip. Solenoid sets booster clutch, engine power actuates trip rope drum. Pushbutton control for trip is recessed in top of swing-lever grip. Nickel-chrome crowd rack is drop-forged. Buckets have detachable drop-forged teeth.

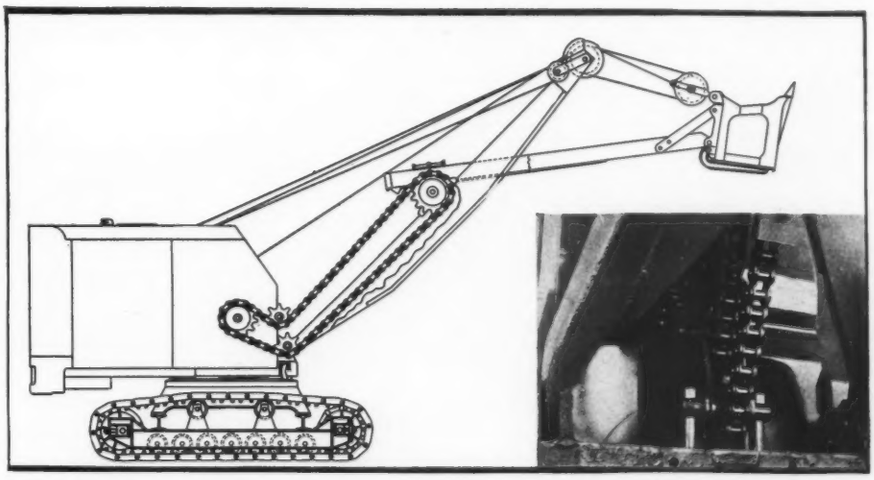




Swing lock mounted on revolving table is operated by foot lever from inside cab. It engages large ring gear in base casting and positively locks table and cab in any position of the circle.

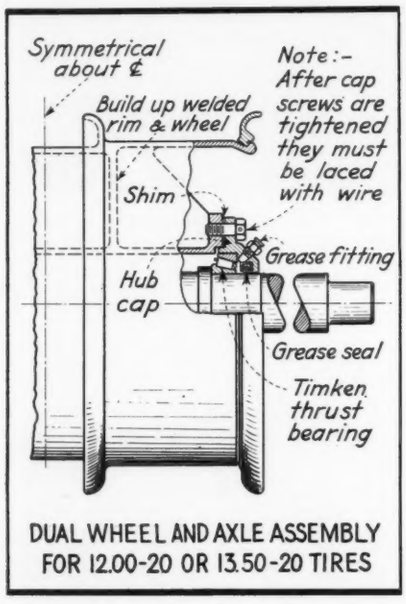
Crawlers are supported on welded box frame. Large-diameter drop-forged rollers are inserted within the crawler frame and ride between special hardened dual lugs on crawler shoes. Drop-forged and heat-treated crawler shoes have drilled pin holes fitted with alloy steel pins. Crawler sprockets are hardened.

Chain crowd. Continuous chain runs from the forward drum to shipper shaft. Automatic chain adjustment, spring loaded, maintains proper tension at any boom angle, eliminates adjustment of chain length. Chain is more efficient and less costly to maintain than cable.

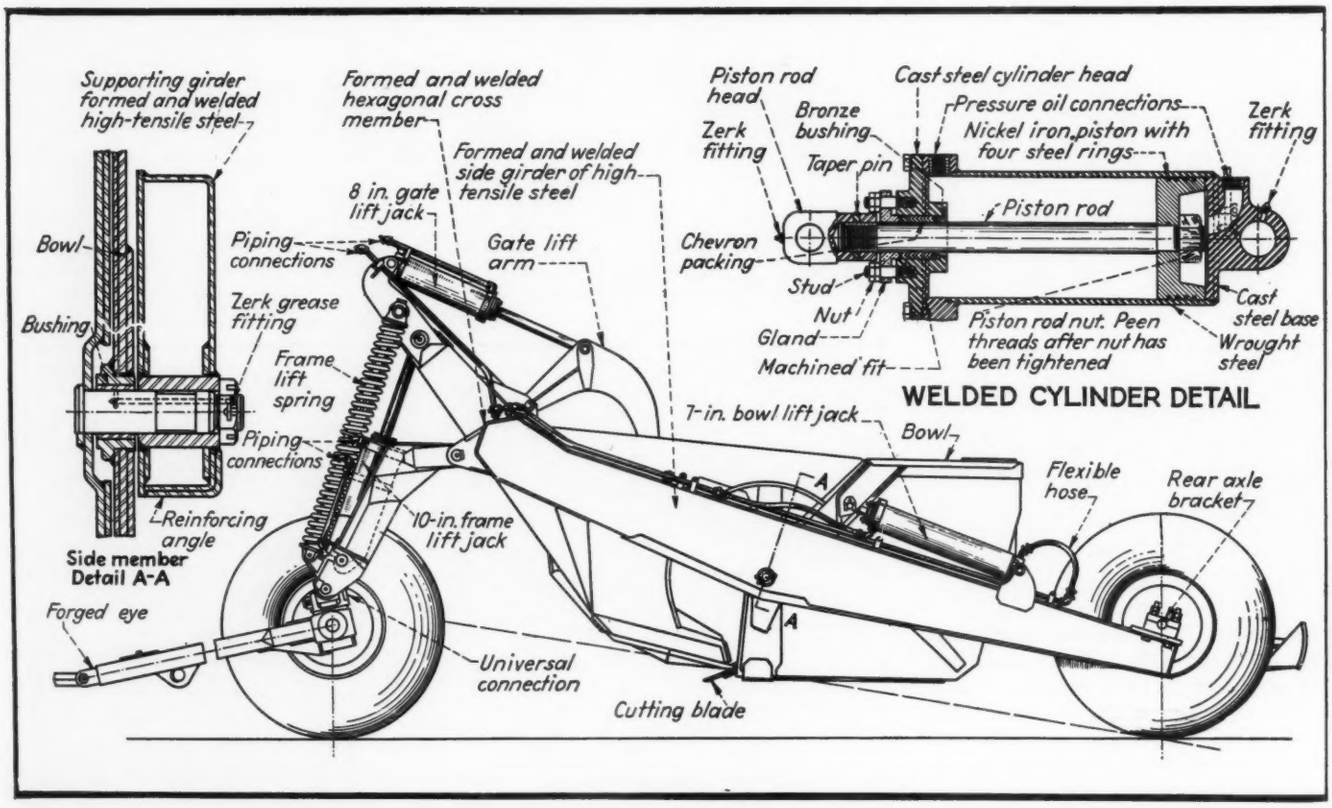


Alloy Steels Reduce Weight, Increase Payload

FORMED AND WELDED high-tensile alloy steel sheets make up the bowl and supporting framework of Gar Wood hydraulic scrapers. Three hydraulic cylinders, supplied with pressure oil from a centralized hydraulic control unit attached to power take-off of tractor, control cut, rate of discharge, depth of spread and positive gate action. Replaceable bowl runners are manganese steel.


Symmetrical about ℄
Note:- After cap screws are tightened they must be laced with wire
Build up welded rim & wheel
Shim
Grease fitting
Hub cap
Grease seal
Timken thrust bearing
DUAL WHEEL AND AXLE ASSEMBLY FOR 12.00-20 OR 13.50-20 TIRES

Main side members are channels of varying depth, formed from a single sheet; no cover plates are required. Front cross member, formerly tubular, is now a hexagonal section formed in two parts and assembled by welding. Main hydraulic lines are inclosed and protected in this member; this was not economically possible when a steel tube was used. Further, the manufacturer need only stock steel sheets. Spring lift mechanism is compressed when blade is in digging position. As load increases, operator opens blade control valve slightly and springs snap cutting edge upward reducing cutting depth and relieving drag on tractor. When fully loaded, blade control release causes spring action to lift bowl to carrying position. In this position scraper wheelbase is shortened aiding maneuverability and permitting faster hauling. Extended cutting edge eliminates side bowl friction and cuts loading time. Cylinder heads of cast steel now have a machined fit to wrought steel cylinder.



News and Summaries

Scientists Using New Atom Smasher Find Precise Voltage Standards

Dr. E. U. Condon, noted Westinghouse nuclear physicist, reports new electrical yardstick and answers question "Why atom smashing?"

SCIENCE'S "BLITZKRIEG" bombardment of the atom has resulted in the discovery of a new and more precise yardstick for measuring high voltages, reveals nuclear physicist Dr. E. U. Condon, Associate Director of Westinghouse Research Laboratories. Using their new electrostatic generator as a cannon for shooting protons (hearts of hydrogen atoms) into the cores of three different elements, Dr. Condon, Dr. W. H. Wells and their colleagues have not only changed one metal into another but have produced new and more accurate measurements of the masses of atoms and high voltage standards by means of neutron emissions.

When a lithium atom suffered a direct hit from a proton, it gave up one neutron and changed into a beryllium atom. Similarly, carbon atoms changed to nitrogen atoms, and atoms of beryllium changed to boron atoms.

All of these reactions occur at a very sharp and definable voltage, reportedly more abrupt than those for gamma rays which have been used in the past for voltage measurement.

Why Atom Smashing?

When asked by PRODUCT ENGINEERING, "Why atom smashing?", Dr. Condon replied: "We cannot ask the engineer to produce useful applications of phenomena that he has never heard of —of phenomena as yet unknown to any man. For continued progress our civilization must support the work of men who will study the fundamental laws of energy and matter as clearly and thoroughly as possible. After the knowledge is gained it is time enough to start thinking about how to use it.

"Accordingly we need to follow up vigorously each new road to knowledge. A few years ago the first steps to development of a new science were taken —the new science was nuclear physics, popularly known as "atom smashing." From this work we already know vastly more about matter than we knew ten years ago. We know two kinds of particles, the positron and the neutron,— fundamental constituents of all materials—the existence of which wasn't suspected a decade ago. We now know how to make nearly all chemical substances behave like radium—instead of having only a few naturally occurring elements at our disposal which show this property. We now know a great deal about the forces which hold atoms together and many other things totally unknown ten years ago. All this new knowledge deepens our understanding of nature profoundly.

"Today we are pushing forward on a new front unaware as yet of the exact

Pear-shaped 90-ton cannon recently completed at Westinghouse mobilizes 4,000,000 volts to shoot sub-atomic particles at metallic targets. Atom-smashing particles hurtle down its 40-ft. vacuum tube at speeds approaching 100,000,000 mi. per hr.

Dr. W. E. Stephens and Dr. E. U. Condon use this cloud chamber for studying sub-atomic collisions caused by the atom smasher. When the air in the chamber is suddenly expanded, moisture condenses about charged particles, making them visible

uses to which this new knowledge will be put. But already there are interesting indications—neutrons may find special applications in place of X-rays for certain problems in cancer therapy—so may also some of the new artificial radio-active substances. Also these new radio-active materials permit new advances in chemistry, especially in the chemistry of our bodily processes, which in turn may aid the advance of medicine.

"If we look back it is hard to find a principle in physics or chemistry the knowledge of which has not found a useful application. This being so we work confidently at the new frontiers turning up the new facts on which will be founded the technical progress of the future."

Light-Weight Diesels Perfected for Aircraft

THE DIESEL ENGINE is "a logical power plant for aviation," asserted Paul H. Wilkinson before the Society of Automotive Engineers at the annual meeting held in Detroit, January 15-19. Modern light-weight, high-speed diesel performance is fast approaching that of comparable gasoline airplane engines, a fact which has led to increasing acceptance of the diesel in European aircraft.

Tabulating the characteristics of gasoline and diesel aircraft engines for comparison, Mr. Wilkinson showed that in all power classes not only can parity be attained, but in many instances the specific fuel consumption of the diesel is much lower.

Mr. Wilkinson, in his observations abroad and at home, has found that diesel-engined airplanes have started easily, run smoothly, accelerated quickly and did not vibrate or smoke under normal conditions. Additional advantages of diesel over gasoline engines are reduced fire hazard and large payload and flight range possibilities. As an instance of the fuel economy of the diesel, Mr. Wilkinson cited the Junkers 86K bomber, powered with Jumo 205 diesel engines. This bomber, when compared with a gasoline-powered plane of the same class, saves 100 lb. of fuel per hour.

A great deal of attention is being paid to details of diesel engine design in this country, as evidenced by a number of other papers presented.

Research Simplifies Design of Diaphragms

CORRUGATED metallic diaphragms have always been selected by trial and rejection, but research recently conducted by the Bureau of Standards might lead to precise design methods. The data prepared by the Bureau takes a step in the right direction by comparing certain types of diaphragms and presenting formulas helpful in the design of similar types.

Diaphragms, singly or in pairs as capsules, find a wide use in pressure measuring instruments. In practice, certain limitations confront the designer who wishes to make use of diaphragms. For one thing, there are no rational design formulas by which the behavior or performance of corrugated diaphragms can be predicted. Even for the limiting case of flat disks, convenient charts, tables, or other aids to design are not yet available. Corrugated disks make possible larger deflections, and afford a control of the pressure-deflection curve.

To obtain data with which more precise design might be achieved, a large number of corrugated diaphragms having geometrically similar outlines were formed by hydraulic pressing. The diaphragms were made of beryllium-copper, phosphor-bronze, and Z-nickel in various diameters and thicknesses. The relations between measured pressures and deflections of the diaphragms were correlated by dimensional analysis to obtain formulas and charts applicable to the design of similar diaphragms. These data were compared with those available on flat diaphragms, and on diaphragms with different types of corrugation.

Newly developed method for bonding soft or synthetic rubber to metal was used in applying a 1-in. rubber coat to this steel pickling roll. In the Hewitt Dura-Bond process, chemically modified rubber is vulcanized to metal, with adhesion strength of 500 to 700 lb. per sq. in. Brass plating and hard rubber base are eliminated.

Meetings

AMERICAN SOCIETY OF TOOL ENGINEERS—Annual meeting, New York, N. Y., March 7–9. Ford R. Lamb, secretary, 2567 W. Grand Blvd., Detroit, Mich.

SOCIETY OF AUTOMOTIVE ENGINEERS—National aeronautic meeting, Washington Hotel, Washington, D. C., March 14–15. John A. C. Warner, secretary, 29 W. 39th St., New York, N. Y.

AMERICAN FOUNDRYMEN'S ASSOCIATION—Annual meeting, Palmer House, Chicago, Ill., May 3–10. C. E. Hoyt, vice president, 222 W. Adams St., Chicago.

V-Belt Industry Revises Power Ratings

POWER RATINGS of multiple V-belt drives have been revised to provide more uniform belt life. Seeking the longest life with the lowest initial cost, engineers of many V-belt manufacturers working in cooperation have standardized sheave diameters and correction factors according to formulas based on results obtained with multiple drives over a

period of ten years. Overload factors also are being arranged to allow for the type of prime mover and driven machine. The new ratings provide in many cases a more compact drive.

Lincoln Foundation Launches Program

WITH the announcement of their 2½-year Industrial Progress Award Program, 458 awards ranging from the Grand Award of $13,700 down to 223 Honorable Mention Awards of $100 each are established by the James F. Lincoln Arc Welding Foundation for studies bringing out benefits of a social, scientific, economic, or commercial nature which can be attributed to the application of the electric arc process of welding during the coming two and one-half years.

Any type of progress which creates benefits of a social, commercial, economic or industrial nature may be brought out in a study. The contest is not simply one to stimulate studies in arc welding, but is being conducted to encourage the type of study which is always productive of great benefits.

Innumerable subjects are suitable for consideration. The twelve classifications and 46 divisions of participation cover every conceivable machine, building, structure, or product made of ferrous or non-ferrous metal. The program is open to everyone who plays any part in actually bringing about progress in the executive, design, manufacture, construction, or maintenance phase of industrial product or structure development.

Do You Know That—

A CURRENT-DETECTING DEVICE has been created which detects currents as low as $1x10^{-21}$ amp.—a current equal to a single electron passing through a wire every five minutes. (4)

PORCELAIN ENAMEL protective coatings have recently been applied, inside and out, to welded steel hot water tanks. Corrugated steel roofing, enameled on both sides, is also new. (5)

CORROSION PITTING of polished sheet aluminum has been traced to the steel wool used in polishing. Electrolytic action between minute particles of iron and iron chloride was attacking the aluminum. Solution was to use stainless steel sponges. (6)

Case Histories In Patent Law

H. A. TOULMIN, JR.

(EDITOR'S NOTE: *We wish to emphasize that in consenting to furnish this column wherein general explanations of the law in specific patent cases are set forth, Col. H. A. Toulmin, Jr., wants it clearly understood "that each actual case should be submitted to your own patent attorney who is the only one familiar enough to advise you accurately." The purpose of these explanations is to give a better general understanding of questions of patent law.*)

No. 14

QUESTION: Can prices and uses of patented articles be controlled?

ANSWER: The Supreme Court of the United States held some years ago that a manufacturer, or owner of a patent, could, under the protection of a patent, control the price at which the licensee under the patent sold the patented article. It has also been held that the licensee can be controlled as to the uses to which he puts the article, and if he sells such an article to any others who use the patented apparatus for a purpose other than that for which it was licensed, that such other persons will become infringers. These two companion rules are illustrated in two Supreme Court cases.

In United States v. General Electric, 272 U. S. 476, the Supreme Court of the United States said:

". . . the patentee may grant a license to make, use and vend articles under the specifications of his patent for any royalty or upon any condition the performance of which is reasonably within the reward which the patentee by the grant of the patent is entitled to secure."

". . . If the patentee . . . licenses the selling of the articles, may he limit the selling by limiting the method of sale and the price? We think he may do so, provided the conditions of sale are normally and reasonably adapted to secure pecuniary reward for the patentee's monopoly. One of the valuable elements of the exclusive right of a patentee is to acquire profit by the price at which the article is sold. The higher the price, the greater the profit, unless it is prohibitory. When the patentee licenses another to make and vend, and retains the right to continue to make and vend on his own account, the price at which his licensee will sell will necessarily affect the price at which he can sell his own patented goods. It would seem entirely reasonable that he should say to the licensee, 'Yes, you may make and sell articles under my patent, but not so as to destroy the profit that I wish to obtain by making them and selling them myself.'" (P. 490)

The Court further said that ". . . price fixing is usually the essence of that which secures proper reward to the patentee." (P. 493)

General Talking Pictures Corporation v. Western Electric Co., 305 U. S. 124, involved amplifiers which were useful in the commercial field and in the private or home field. Western Electric was granted an exclusive license in the commercial field. The American Transformer Co. was one of 50 non-exclusive licensees who were restricted to manufacturing and selling these amplifiers for use in the private and home field only. The Court said:

"Transformer Company, knowing that it had not been licensed to manufacture or to sell amplifiers for use in theatres as part of talking picture equipment, made for that commercial use the amplifiers in controversy and sold them to Pictures Corporation for that commercial use. Pictures Corporation ordered the amplifiers and purchased them knowing that Transformer Company had not been licensed to make or sell them for such use in theatres. Any use beyond the valid terms of a license is, of course an infringement of a patent. . . . If where a patented invention is applicable to different uses, the owner of the patent may legally restrict a licensee to a particular field and exclude him from others, Transformer Company was guilty of an infringement when it made the amplifiers for, and sold them to, Pictures Corporation. And as Pictures Corporation ordered, purchased and leased them knowing the facts, it also was an infringer." (P. 126)

"The question of law requiring decision is whether the restriction in the license is to be given effect. That a restrictive license is legal seems clear. . . . The practice of granting licenses for a restricted use is an old one. . . . So far as appears, its legality has never been questioned." (P. 127)

"As the restriction was legal and the amplifiers were made and sold outside the scope of the license the effect is precisely the same as if no license whatsoever had been granted to Transformer Company. And as Pictures Corporation knew the facts, it is in no better position than if it had manufactured the amplifiers itself without a license. It is liable because it has used the invention without license to do so." (P. 127)

Question and Comment

Simplified Formula for Cam Rise and Fall

H. A. BOWER

In calculations to determine the rise or fall of a follower for a given angle of rotation of a cylindrical eccentric cam, the amount of eccentricity is usually multiplied by the versed sine of the angle. Expressing this as a formula,

$$C = R \; versin \; \theta \qquad (1)$$

This states that the rise or fall of the follower is equal to the vertical displacement of the cam center as it revolves about the fixed drive center, through the angle θ. Except where a fine degree of accuracy is desired, this is true.

A simple test, whereby it may be determined whether the results of this simpler formula come within the limits of desired accuracy is to equate:

$$E = R^2/2A$$

This value of E is the maximum amount by which the basic equation errs, and occurs only when the angle $\theta = 90$ deg. If this value of E is found to be so small as to be within the desired limits, the equation $C = R \; versin \; \theta$ may be used for either cam rise or cam fall.

However, if the error is greater than the limits of accuracy, it must be mathematically computed and included in the formulas for cam rise and fall. As either diagram shows, when the cam is rotated away from the high or low point, its center is no longer on the line between the centers of the drive and of the follower. The point of contact of cam and follower, remaining on the line contacting their centers, is displaced from the vertical with the result that there is a slight gap between the cam and follower on the vertical center line. Because of this gap, the value of C differs from $R \times versin \; \theta$ by some value of error, E. In the diagram the proportions are exaggerated so that the gap can be seen. From Fig. 1,

$$C + A = R \; versin \; \theta + (A - E)$$

or,

$$C = R \; versin \; \theta - E$$

Since θ is measured from the low point of the cam, the expression is the skeleton formula for cam rise. In Fig. 2, the angle is measured from the high point of the cam, and we find

$$C = R \; versin \; \theta + E$$

the skeleton formula for cam fall.

It can be shown that these formulas apply to any angle of cam rotation up to 180 deg. from the high or low point when the proper value of the versed sine is used.

The angle θ on the diagram shows the displacement of the contact point from the vertical. It will be shown that the value of this angle need not be determined except in the development of the formulas. Using either figure,

$$cos \; \phi = (A - E)/A$$

$$E = A - A \; cos \; \phi = A \; (1 - cos \; \phi)$$

$$E = A \; versin \; \phi$$

$$sin \; \phi = \frac{R \; sin \; \theta}{A}$$

therefore

$$E = A \; versin \; sin^{-1}\left(\frac{R \; sin \; \theta}{A}\right)$$

Substituting for E in the skeleton formulas,

$$Cam \; rise = R \; versin \; \theta - A \; versin \; sin^{-1}\left(\frac{R \; sin \; \theta}{A}\right) \qquad (2)$$

$$Cam \; fall = R \; versin \; \theta + A \; versin \; sin^{-1}\left(\frac{R \; sin \; \theta}{A}\right) \qquad (3)$$

Another pair sometimes used:

$$Cam \; rise = R \; versin \; \theta - A + \sqrt{A^2 - R^2 \; sin^2 \; \theta} \qquad (4)$$

$$Cam \; fall = R \; versin \; \theta + A - \sqrt{A^2 - R^2 \; sin^2 \; \theta} \qquad (5)$$

Since trigonometry tables are used in any event, Equations (2) and (3) are to be preferred, as far less work is required for the same degree of accuracy. From differential calculus

$$(versin \; \phi)' = \tfrac{1}{2} \; sin^2 \; \phi$$

for angles from 0 to 90 deg. When the value of ϕ is very small, as is usually the case in eccentric cam work, it is found that the versed sine of ϕ is equal to half the square of the sine of ϕ, and one may be substituted for the other in an equation. Thus for $A \; versin \; sin^{-1} \; (R \; sin \; \theta/A)$, substitute $A \; \frac{(R \; sin \; \theta/A)^2}{2}$ or $\frac{R^2 \; sin^2 \; \theta}{2A}$

Equations (2) and (3) become:

$$Cam \; rise = R \; versin \; \theta - \frac{R^2 \; sin^2 \; \theta}{2A} \qquad (6)$$

$$Cam \; fall = R \; versin \; \theta + \frac{R^2 \; sin^2 \; \theta}{2A} \qquad (7)$$

Should R have a value as high as one-tenth that of A, the short-cut Equation (1) introduces an error the maximum value of which is 0.0000126 R, and if the ratio of R to A is such that ϕ does not exceed 3 deg., the error cannot exceed one millionth of R.

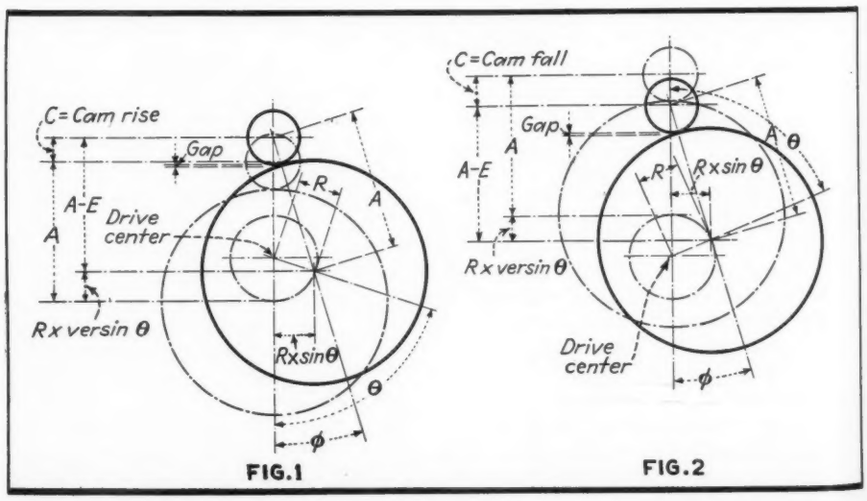


FIG.1 FIG.2

V. D. I. Publication Gives Steam Jet Design Data

To the Editor:

Data on the design of steam jet ejectors, requested by Mr. A. Cunningham on page 36 of the January number, does exist in a German publication.

The V. D. I. publication *Vorshungsheft 395* gives a beautiful derivation of

all the equations necessary for design of injectors, eductors and ejectors for fluids and gases, homogeneous or heterogeneous. I have been able to use the material several times myself, and recommend it to Mr. Cunningham.

—J. R. HARKNESS
Denver, Colo.

Variable-Speed Hydraulic Coupling

To the Editor:

On page 33 of your January number, in the article "Motor Drives for Vertical Shafts—II," by H. D. Seymour, there is shown (Fig. 8) an application of a variable-speed hydraulic coupling. We are interested in this coupling, and would greatly appreciate your sending us further information concerning it.

—W. C. JEROME
Worcester, Mass.

[*Editor's Note*—That same variable-speed hydraulic coupling is described, with a sectional view to show how it operates, in the January number, page 10. Hydraulic Coupling Division, American Blower Corporation, Detroit, can furnish further details. The drive shown by Mr. Seymour is one of the first applications of this coupling.]

Error in Formula For Cantilever Beam

To the Editor:

Equation 6 on page 18 of the January number, in my article "Cantilever Beams," is in error. The equation is an expression for the deflection Δ and not for x. Therefore, the x on the left hand side of the equals sign should read Δ.

I would like to take this opporunity to compliment PRODUCT ENGINEERING on the pleasing appearance and the manner in which the article was presented.

—C. C. STREET
Federal Products Corporation

[*Editor's Note*—Don't blame Mr. Street for this error. Since we own the pencil that should have (but didn't) put a Δ in the blank space left by a typewriter which couldn't type a Δ, we'll have to take the blame.]

Well, Well!

To the Editor:

Mr. H. E. Smith's January "Brain Tester" about the pebble falling into the well drilled through the center of the earth to China suggests some intriguing situations. I admit I don't know much about celestial mechanics, but I'm sticking my neck out anyway.

It is quite obvious that the pebble would strike the east wall of the well. This can be explained by the fact that when the pebble is released, it is a freely moving body with an initial tangential velocity equal to that of the surface of the earth at the opening of the well. As the pebble falls toward the center of the earth, the walls around it naturally move at slower and slower tangential speeds, so the pebble will catch up to the east wall before it has fallen far.

But, let us take the supposition that the earth is a hollow shell, with its mass concentrated at its center. Then what would be the path of the pebble? As nearly as I can figure out, the path would be a spiral, winding up in a circular path of such a radius that the centrifugal force is equal to the force of gravity. This doesn't seem logical until it is considered that the pebble, in falling, attains a very high velocity.

With the picture of this path in mind, wouldn't it be possible to have a well of such a diameter that the pebble would not strike its wall?

A factor which must be considered is that the force of gravity diminishes in a straight line ratio, according to Bowser, as the pebble approaches the earth's center. One peculiar effect of this, not related to our problem, occurs if we assume that the earth is not rotating. Then it would take a pebble about 21 min. to reach the center, and it doesn't make any difference from what point, at or beneath the surface of the earth, the pebble is released.

Here is another tricky possibility. Let us suppose, with the spiral path again in mind and assuming that the gravity force is concentrated at the earth's center, that the pebble is a very large boulder. As the boulder approaches the radius where planetary motion occurs, the force of gravity acting on the inner half of the boulder exceeds the centrifugal force acting on that half, and vice versa for the outer half. Therefore, although the sum of these forces is zero, there is a force tending to pull the boulder apart. If the boulder is large enough, the stresses set up by this force would cause the boulder to explode.

These are logical conclusions, at least to me. I have no doubt that the fundamentals and formulas of these cases have all been worked out and are well known to the savants. Some day I'll look them up.

—WILLIAM FERRY
New Haven, Conn.

Can You Work This One?

H. E. SMITH

This month's problem—

A Matter of Course

A British gunboat, traveling at 25 mi. per hr., was pursuing an enemy merchant vessel which traveled at 15 mi. per hr. The gunboat, steadily overtaking the freighter, was exactly 5 mi. away from it when a heavy fog descended. Immediately the merchantman, taking advantage of the fog, changed his direction. His course from that point on was a straight line.

The captain of the gunboat did not wish to give up the chase. He knew the speed of each of the two boats, and their distance apart, and he was quite sure that the other boat would go off in a straight line at some unknown angle. From these facts, he was able to sit down and plot a course to follow by which he could be certain of intercepting the freighter.

What was the course he plotted?

Solution to January problem—

Brain Testers

1. If a straight well could be bored through the center of the earth to China and a pebble dropped from the exact center of the well at the surface of the earth, the pebble would strike the east wall of the well. This is because as the pebble moves toward the center of the earth, its tangential velocity remains constant while the tangential velocity in the eastward direction at different cross-sections of the tube diminishes to zero at the center of the earth.

2. Where a bottle and a cork cost $1.05, the bottle costing $1.00 more than the cork, the cork must have cost 2½ cents.

3. The number 105,263,157,894,736,842 can be exactly doubled by moving the digit 2 from the right-hand end to the left-hand end of the number. This number can easily be arrived at by starting at the last digit, 2, multiplying it by 2 to give the next digit 4; multiplying 4 by 2 to give 8; multiplying 8 by 2 to give the digit 6 with one to carry, and so forth until the ultimate digit 1 is produced.

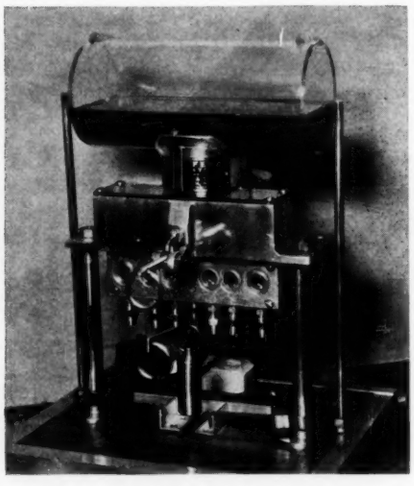
4. If 3,001 digits are required to number the pages of a book, the first 999 pages require $9 + 180 + 2{,}700 = 2{,}889$ digits. 112 additional digits are employed, which suffice to number 28 more pages. The book has 1,027 pages.

5. The fish that weighed 10 lb. plus half his weight weighed 20 lb., of course!

New Materials and Parts

Automatic Lubricator

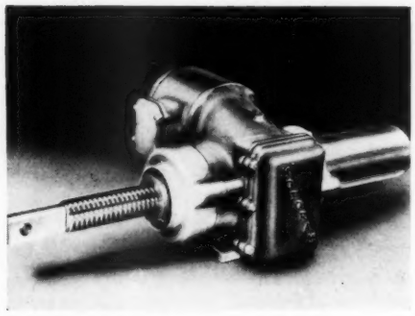
Mykrofeed automatic lubricator utilizes controlled gravity flow in supplying oil to machine bearings and other points of lubrication. When the machine is started, a solenoid opens a ball-valve in the transparent plastic reservoir, releasing oil into a constant-level chamber. The reservoir can also be opened by hand. Feed rates to individual outlets are controlled by capillary action of wicks which lie in an oil trough fed from the constant-level chamber. When the reservoir is closed by shutting off the current to the machine, an automatic-



ally decreasing supply of oil is furnished by the wick units to the bearings for some time, eliminating the danger of dry bearings upon restarting the machine. Provision is made for flood-feed of bearings after a long shut-down. Major parts are of aluminum. Transparent plastic cover and sight feed fronts provide visibility of operation. Multiple distributor feed units are available for attachment to the Mykrofeed. Bromley & Son, 232 Chestnut St., Kearny, N. J.

Motoram

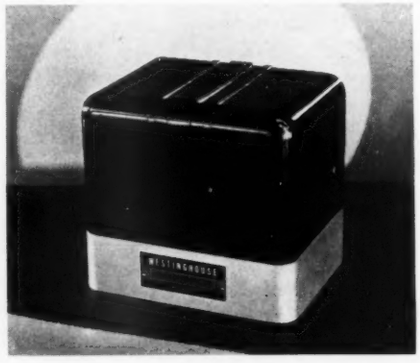
Designed for application on doors, clutches, valves, variable speed drives, or any other applications requiring a straight line power stroke in either or both directions, this new mechanism is operated by an electric motor. Self-contained electrical and mechanical limit



switches stop the movements of the shaft at any predetermined point of travel in either direction and sustain a load on the shaft after the motor is stopped. The movable thrust shaft has a minimum travel of ½ in. and a maximum of 72 in. or more. Movement may be automatically stopped or reversed under any one of the following conditions: predetermined thrust in both directions and of different capacities; thrust in one direction and definite distance of travel in the opposite direction; definite movement in either direction set to stop within 0.005 in.; accidental overload under any conditions of stopping. Power is imparted to the shaft in both directions, eliminating return spring. By the use of any standard time clock, the Motoram may be operated through time intervals. Four sizes of units are available with capacities of 100 lb. to 3,000 lb. thrust at any speed of travel. Electromarine Corp., Westport, Conn.

Photoelectric Relay

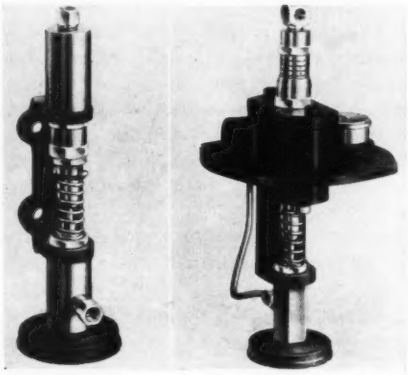
Compact photoelectric relay, designated as Type RQ Photo-Troller, can be used for counting, sorting, weighing, measuring, signalling, and similar functions. The device is especially designed



for low operating expense, low initial cost, and ease of installation. It has two normally open and two normally closed contacts, each rated at 10 amp. on 115-volt a.c. circuits. Operates on a change of 40 foot-candles, if not more than 15 foot-candles of extraneous light is present. Increased sensitivity is obtainable by means of a condensing lens. The unit may be wall- or bench-mounted with open wiring, or with concealed or exposed conduit. It has a gray die-cast chassis with a black molded cover which is removable without tools. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Hydraulically Operated Lubricators

New types of automatic lubricators are driven directly from the hydraulic circuit of hydraulically operated machines, eliminating mechanical drives from machine shafts or moving units. The lubricator cylinder is driven by an hydraulic

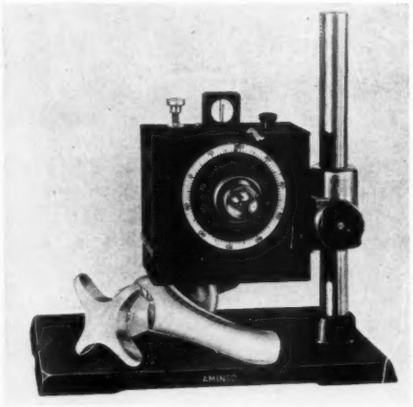


cylinder connected to the hydraulic circuit, the two pistons being connected by a common piston rod which is ball-jointed to each to insure a free working assembly. During the high pressure period, the hydraulic cylinder drives the lubricator pump in one direction, discharging the oil to the bearing distribution system. During the low-pressure period, the assembly is returned by a spring. A simple screw adjustment permits setting the working stroke to any length required to meet the lubrication needs of a particular machine. The pump gives positive control through metered feed. Two types of lubricator pumps are available: Type C for mounting on a standard oil reser-

voir; and Type G for mounting on a reservoir cast integral with the machine. Bijur Lubricating Corp., Long Island City, N. Y.

Instruments for Measuring Thicknesses of Coatings

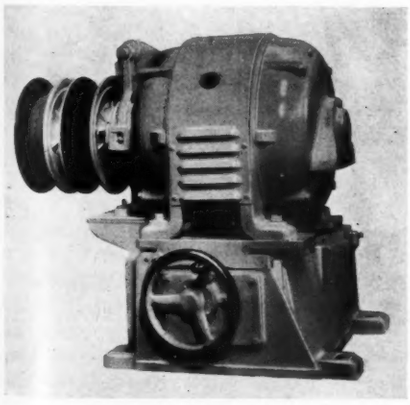
New Magne-Gage, an instrument for measuring local thickness of coatings on metals by the rapid, non-destructive magnetic method, has been improved to greatly increase the utility of the instrument. It is now possible to measure various types and thicknesses of coatings with



only one instrument by interchanging different types of magnets. The instrument will measure nickel coatings on non-magnetic base metals, non-magnetic metallic or organic coatings on magnetic base metals, and nickel coatings on iron or steel. Bulletin PE-2070 available. American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.

Multiple Belt Variable Speed Drive

Speed changes on this new motor unit are effected by turning the handwheel which shifts the motor on the base. This moves the yoke which actuates the multi-



ple pulley. Motors feature new lubrication system. A hollow section of the shaft is constructed through the pulleys and grease is ejected through orifices from the hollow section to the sliding surfaces. Adjustment of the belt may be made without shifting the entire base by turning a set screw which turns the eccentric bushing located at the pivot point of the shifting yoke arm. U. S. Electrical Motors, Inc., 200 E. Slauson Ave., Los Angeles, Calif.

Solenoid Valves

Constituting an improvement over the Type D valves formerly available, this new valve will operate on 250 lb. per sq. in. instead of 150 lb. per sq. in., has a new solenoid with glass insulation which will withstand higher temperatures, and has a new modernistic coil shield which helps to dissipate heat. These valves are designed to handle freon, methyl chloride, sulphur dioxide, brine, air, oil,

water, steam, and ammonia. Available in horizontal or angle, screwed ends, in sizes from ⅛ in. to 3 in. Horizontal or angle, flanged ends, may be had in sizes from ¾ in. to 3 in. Hoppe Engineering Co., Indianapolis, Ind.

Vernier Control for Variable Speed Transmission

Designed for installations where extremely fine control of speed changes is required, this vernier control is available for attachment to all sizes of the Link-Belt P.I.V. Gear variable speed transmission. The control can be supplied with either of two ratios, 7½ to 1 or 30 to 1, and is equipped with two hand wheels. One hand wheel is for direct control; the secondary vernier control wheel will turn either 30 or 7½

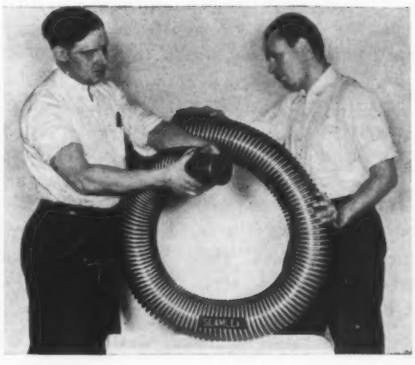
revolutions to one of the direct wheel, depending upon which ratio of worm-gear reduction is furnished. Link-Belt Co., 2045 W. Hunting Park Ave., Philadelphia, Pa.

Protective Wax

Designed to protect the production finish on anodic treated aluminum during handling, shipping, and erection, this liquid material is applied with a brush or cloth. It is allowed to remain on the surface until after fabrication, at which time it can be rubbed down. It leaves a firm transparent wax surface which serves as protection against discoloration. Production Wax No. 2 is approved for use on all types of aluminum surfaces that have been given an anodic treatment. It is also suitable for use on bronze, chrome and other bright surfaces. Skybryte Co., 3125 Perkins Ave., Cleveland, Ohio.

Seamless Metal Hose

Pronounced S-shaped corrugations ¾ in. deep and with relatively short pitch, are responsible for the unusual flexibility of this 4-in. seamless bronze tube which is shown bent to a radius of 10 in. The convolutions form a left-hand helical thread, assuring uniform distribution of stress and therefore long life of hose. Covered with bronze braid jacket, it will



withstand a safe internal working pressure of 150 lb. per sq. in. Suitable for conveying steam, water, oil, gas, air, etc. Weight per foot is 6.5 lb. Pipe thread coupling or flanges are available in standard sizes. Seamlex Co., Inc., 5-17 48th Ave., Long Island City, N. Y.

Industrial Tire

Combining the advantages of both the pneumatic and solid type of industrial tire, the new Silvertown Cushion Type tire is light, has high cushioning qualities, streamlined appearance and yet has the sturdy construction and load-carrying capacity of a solid tire. The new design eliminates the hazard of tire failure caused by punctures or leaky valves and requires no servicing such as

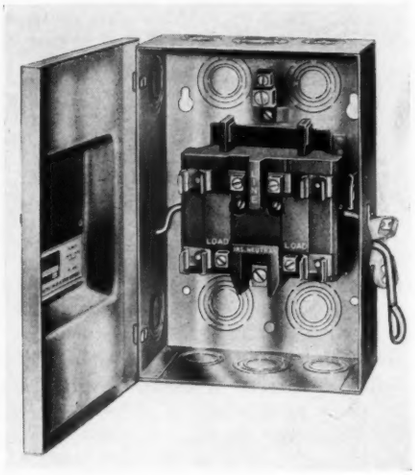
inflation or tire repairs. Interchangeable on the same wheel with single-tube pneumatic tires of like sizes. Available at present in two sizes, 6 x 2.00 and 8 x 2.50. The B. F. Goodrich Co., Akron, Ohio.

Sound Absorption Felt

New absorption felt, Style 1660, is light in weight, pliable, and easily handled, is fire-resistant, and will not gain appreciably in weight from moisture absorption. Its sound absorption coefficients are 0.54 for frequencies in the range of 512 cycles per sec., 0.77 for frequencies of 1,024 cycles per sec., and 0.84 for frequencies of 2,048 cycles per sec. Sound frequencies in the average building range around 512 cycles per sec. Aircraft frequencies range from 1,024 to 2,048 cycles per sec. The felt is claimed to have a low heat conductivity, and is non-corrosive in contact with aluminum and its alloys. The Felters Co., Inc., 210 South St., Boston, Mass.

Safety Switch

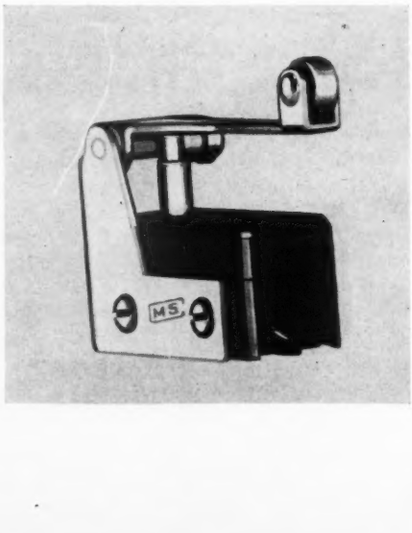
New Type D safety switch is equipped with the Dualbreak mechanism, effectively used in Type A and Type C safety switches. Two separate groups of neutral terminals include one grounded to the cabinet, located at the upper end of



the box for service installations; and the other designed for insulated neutral application, located at the lower end of the switch block. Cabinet is compact, and can also be supplied in rain-tight, weatherproof inclosures of galvannealed steel, with baked aluminum finish. Colt's Patent Fire Arms Mfg. Co., Hartford, Conn.

Attachable Actuator For Micro Switches

This actuator, an attachment to be used in conjunction with regular and Metal-Clad micro switches, is easy to attach, easy to adjust, and has a long life. The level arm of the actuator rides on an oilless bronze bearing which rests on the plunger of the switch. A case-hardened roller with an oilless bronze bearing is mounted on the end of the lever arm. The operating ratio is about



8 to 1. Minor adjustments can be made by slightly bending the lever arm. Micro Switch Corp., Freeport, Ill.

Koroseal Paint

Designated as No. 495 Korolac, the name covering solutions of the plasticized poly-vinyl chloride, this new paint gives a semi-glossy black finish and is recommended wherever extremely corrosive conditions disqualify any other kind of paint or coating. When thoroughly dry, it will withstand all acids, alkalies and salts in the concentrations commonly met with in industry, at temperatures up to 150 deg. F. It is not affected by chromium, nickel, cadmium, zinc, copper, brass, silver or tin plating solutions, nor are such solutions contaminated or fouled by the paint when it is thoroughly dried. The dried film is hard and resistant to abrasion, yet sufficiently elastic to conform to contraction and expansion of the base metal. A Korolac primer for use with the new paint has also been introduced. Both the primer and the paint can be applied by brushing and spraying. B. F. Goodrich Co., Akron, Ohio.

Dynamotors

Precision built dynamotors, designed especial for aircraft, police, marine and amateur transmitters and receivers, are claimed to be extremely light in weight per watt output. Modern, compact design, and sturdy construction contribute to smooth, continuous perform-

ance and long operating life. Starting current is low. Available in a wide range of sizes. Eicor, Inc., 515 S. Laflin St., Chicago, Ill.

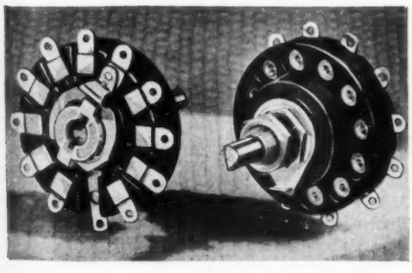
Wheels

Designed to overcome the disadvantages of steel rim wheels, new pneumatic and semi-pneumatic tired spoke wheels are for use on portable tools, sprayers, pushcarts and numerous similar items. Oval type spokes give

strength against side strain. Cast semi-steel hubs are assembled in the wheel with two pressed steel disks driven tight on the hub and riveted together. Both the pneumatic and semi-pneumatic tires have fabric base to prevent slipping and stretching. Geneva Metal Wheel Co., Geneva, Ohio.

High-Current Tap Switch

Small size, open-type, high-current Model 111, like the larger Ohmite tap switches, is a multi-point, load-break, non-shorting, single-pole rotary selector, particularly designed for a.c. circuits. It is all-ceramic in construction, compact, and well insulated. Silver-to-silver contacts are self-cleaning and require no maintenance. Positive cam-and-roller mechanism provides desirable slow-break, quick-make action, minimizes sparking, and increase the contact life.



Ideal for such applications as switchboards, oven controls, welders, X-ray equipment, motor control, pyrometer circuits, etc. Available with a maximum number of eleven taps. Shaft diameter ¼ in. Switches are rated 5 amp. at 120 volts, and 3 amp. at 240 volts, in a.c. circuits. Ohmite Manufacturing Co., 4835 Flourney St., Chicago, Ill.

Manufacturers' Publications

Materials

BEARINGS—Johnson Bronze Co., New Castle, Pa. Catalog L-2, 22 pages, 8½x11 in. New size listing of Ledaloyl self-lubricating bearings, giving more than 850 stock sizes. Considerable technical and application data is included.

CERAMICS—American Lava Corp., Chattanooga, Tenn. "Designing Die Pressed Steatite Ceramics," 4 pages, 8½x11 in. Information needed in the design of die pressed parts is given in this well prepared article. Particular emphasis is placed on the design of parts to effect greater die economies.

MEEHANITE—Meehanite Research Institute of America, Inc., 311 Ross St., Pittsburgh, Pa. Bulletin 10, 12 pages, 8½x11 in. Five principal reasons why Meehanite is used extensively for pressure castings are described in this booklet, which includes a table of pressure-temperature ratings made from tests on valves.

MONEL METAL—International Nickel Co., Inc., 67 Wall St., New York, N. Y. Bulletin T-9, 16 pages, 8½x11 in. The engineering properties of "K" Monel are covered in this technical bulletin, which gives the characteristic of the metal under all conditions.

STEEL—The Babcock & Wilcox Tube Co., Beaver Falls, Pa. Bulletin 12-A, 6 pages, 8½x11 in. Condensed technical data on high temperature steels, containing revised useful information and technical data on seamless alloy tubes and pipes for high pressure and high temperature services.

Mechanical Parts

CONE-DRIVES—Cone Worm Gear Div., Michigan Tool Co., Detroit, Mich. Bulletin, 26 pages, 8½x11 in. The design of Cone Drive worm gearing is covered, giving complete data on ratings, sizes, and ratios, and including engineering formulas for all factors of Cone Drive design.

FLEXIBLE COUPLINGS—Farrel-Birmingham Co., Inc., Ansonia, Conn. Catalog 443, 44 pages, 8½x11 in. Explains the function of a flexible coupling and describes, gives engineering data and illustrates the various types of Farrel flexible couplings.

FLEXIBLE TUBING—Chicago Metal Hose Corp., Maywood, Ill. Bulletin SS-16, 4 pages 8½x11 in. Engineering data on Rex-Bellows stainless steel flexible tubing. Both divided and fully corrugated forms are shown, together with couplings and the new resistance welded pipe joints.

ROLLER BEARINGS — Schaeffer Bearing Corp., 35 E. Wacker Drive, Chicago, Ill. Catalog 15, 86 pages, 8½x11 in. Illustrates and describes complete line of Schaeffer roller bearings and includes recent designs. Of particular interest to engineers are 7 pages of engineering data.

SPEED REDUCERS—The Adams Co., 111 E. 6th St., Dubuque, Iowa. Catalog 201, 12 pages, 8½x11 in. Describes and illustrates MultiSpeed reducers, including the new Type MD, recommended for converting machines from belt to direct motor drive. Speed and horsepower ratings, and dimensions are given.

VARIABLE TRANSMISSION—Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill. Book 1574, 40 pages, 8½x11 in. Covers vernier and other types of control for all sizes of the Link-Belt P.I.V. Gear variable speed transmission, for installations where fine control of speed changes is required.

Electrical Parts

CAPACITORS—Aerovox Corp., New Bedford, Mass. Bulletin, 16 pages, 8½x11 in. In addition to a catalog listing of Aerovox industrial capacitors, this bulletin contains several pages of engineering information pertaining to the selection of an electrolytic motor starting capacitor.

MOTOR CONTROL—General Electric Co., Schenectady, N. Y. Bulletin GEA-1724B, 8 pages, 8x10½ in. Describes and illustrates the new SCI relay for synchronous motor control.

Finishes

LOW-BAKE FINISH—Ault & Wiborg Corp., 75 Varick St., New York, N. Y. Folder, 4 pages, 8½x11 in. Describes Polymerin-100, the new speed-bake finish that makes possible curing temperatures which are often as much as 100 deg. F. below those of the original Polymerin.

HIGH-SPEED ENAMELS—Maas & Waldstein Co., 438 Riverside Ave., Newark, N. J. This technical data bulletin gives information on Codur enamels, including specifications for finishing Bakelite products, steel cabinets and furniture.

Fabrication Methods

SOLDERING—Wayne Chemical Products Co., 9600 Copeland St., Detroit, Mich. Bulletin, 4 pages, 8½x11 in. Describes advantages of Meltomatic paste solder, a new process which is faster and simpler.

WELDING—General Electric Co., Schenectady, N. Y. Bulletin GEA-3313, 8 pages, 8x10½ in. Sets forth the features of the new weld recorder, which records the character of the electrical input for every weld and locks out the welder when the input varies more than the allowable limits.

Books and Bulletins

Diesel Engines Theory and Design

HOWARD E. DEGLER—*62 illustrations, 270 pages. 5½x8¼ in. Brown clothboard covers. Published by American Technical Society, Drexel Ave. at 58th St., Chicago, Ill. Price $2.50.*

Prepared primarily for the use of students, designers and draftsmen who have a knowledge of the principles of mechanics and an acquaintance with the forms of internal-combustion engines, this textbook is concerned chiefly with the application of theory and is supplemented with numerous examples.

Approximately one half of the volume is devoted to the thermodynamics of internal-combustion cycles, fuels, combustion, combustion chambers, testing and performance. This is followed by an exposition and analysis of the principles of engine design, and the design of major engine parts. Only a small amount of descriptive matter covering engine details, operation, and maintenance of diesel engines is presented by Mr. Degler.

Aircraft Design

D. H. LATIMER NEEDHAM—*Two volumes, 5¾x8¾ in. Volume I, 227 pages, 144 illustrations; Volume II, 321 pages, 160 illustrations. Published by Chemical Publishing Co., 148 Lafayette St., New York, N. Y. Price Volume I, $6; Volume II, $6.50.*

Intended for use by engineering students, these books should also be of particular interest to engineers engaged in design and research of airplane structures. The first volume, Aerodynamics, is concerned mainly with principles of flight and stability stated in simple language and extensively illustrated by schematic drawings. After brief introductory chapters devoted to symbols and general principles, applications of fundamental formulas to specific problems of air lift, stability, control are covered in succeeding chapters. This volume also includes a complete chapter on parasite drag.

The second volume, Aerostructures, deals mainly with the mathematical treatment of airplane design, starting with general layout of aircraft, conditions of loading, and following with a discussion of materials suitable for aircraft production. The remainder of the book goes into a comprehensive discussion of the design of each detail of the structure in turn: wing, fuselage, tail unit, undercarriage and control systems. The book concludes with a chapter on metal construction and a chapter on experimental testing.

Based on the author's twenty years' experience in the design, construction and piloting of aircraft these books should be valuable additions to the library of any engineer interested in aeromechanics.

Patent Fundamentals

ADELBERT SCHAPP—*176 pages, 5¾ x 8⅝ in. Maroon clothboard covers. Published by The Industrial Press, 148 Lafayette St., New York, N. Y. Price $2.*

This broad treatise on patents presents to the inventor and engineer a discussion of the best and most approved ways of defining and protecting an invention. Worded in non-technical language, it covers patent fundamentals in logical order, commencing with a clear definition of what is and what is not a patentable invention, and setting forth the precautions every inventor should take from the date of the conception of his invention.

To enable the engineer to better cooperate with his patent attorney, several chapters cover the essentials of preparing applications, developing claims, and interference procedures. The inventor is also told how to protect himself in the assignment of patents and licensing agreements. In conclusion, the author discusses trademarks and copyright protection, as distinguished from patents.

By illustrating his points with numerous practical examples and thorough discussions of many important court decisions, Mr. Schapp has produced a textbook which is easy to read and which, by virtue of its cross-index, can readily be used as a reference.

Number

TOBIAS DANTZIG—*Third Edition. 12 illustrations, 320 pages, 6x8¾ in. Brown clothboard covers. Published by Macmillan Co., 60 Fifth Ave., New York, N. Y. Price $3.*

Mr. Dantzig attempts in his book to restore the popularity of the cultural content present in the mathematical science. The book deals with mathematics, but it deals with ideas back of the symbol or forms, and is one of the few discussions of the subject not cluttered up with involved equations and methods. Fundamental issues are presented without bringing in the whole intricate apparatus of the science.

The third edition contains much historical matter not present in preceding editions. The changes are incorporated in 26 articles added to the text. Some of these deal with history, others with philosophical questions, still others amplify technical points.

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Laminated Phenolic Products N.E.M.A. Standards

Publication 30-57, 28 pages, 8 x 10½ in. Published by National Electrical Manufacturers Association, 155 East 44th St., New York, N. Y. Price 25 cents.

A revision of publication No. 34-26 issued in 1934, this booklet is a reference work of practical information concerning the manufacture, test and performance of laminated phenolic plates, tubing and rods used in the manufacture of electrical apparatus and supplies. It contains a general description of laminated phenolic manufacture; description of twelve grades of plates and nine grades of tubing; standard sizes and finishes of plates, rods and tubing; standard colors of plates; standard tolerances; and physical and electrical properties of the various grades.

Recommended Practice for Machining and Punching Laminated Phenolic Plates

Publication 39-58, 4 pages, 8 x 10½ in. Published by National Electrical Manufacturers Association, 155 East 44th St., New York, N. Y. Price 10 cents.

This new publication includes information on sawing, drilling, tapping, threading, milling, turning, punching, shaving and shearing of the various grades of laminated phenolic plates. Of particular help is a table which lists the comparative punching properties.

Short-Center Pivoted-Motor Leather Belt Drives

64 pages, 8½ x 11 in. Published by American Leather Belting Association, 100 Gold St., New York, N. Y.

Planned and modeled after standard V-belt data books, this book contains all the information needed for the selection of an automatic tension control short-center pivoted-motor leather belt drive. Advantages of the flat leather belt drives are set forth, and results of experiments recently conducted on relative efficiencies of various types of belt drives are given. Engineering data include A.L.B.A. basic horsepower tables, and many pages of tables arranged by horsepower requirements to facilitate the selection of drives.

Typical applications of leather-belt drives are illustrated in a 24-page section of the book, which groups the installations by types of industries, for selective reference.

A. C. Magnet Coils

Design Fundamentals

GRAHAM LEE MOSES
Westinghouse Electric & Manufacturing Company

MAGNET coils for use on a.c. circuits present quite a different problem from those used in d.c. circuits. The difference is largely by reason of "inductance," which sets up an electromotive force that opposes changes in the current. This electrical characteristic corresponds to the physical phenomenon of inertia or "flywheel effect." Just as a flywheel opposes the mechanical forces trying to either accelerate or decelerate it, so does an electrical circuit possessing inductance oppose electromotive forces (voltage) tending to increase or decrease the current.

The presence of inductance in an electric circuit prevents current from instantaneously attaining the maximum value determined by voltage and resistance under Ohm's law. The exact time for the current to reach this maximum is dependent upon the ratio between the inductance and resistance in the circuit. Expressed numerically, the inductance in henries divided by the resistance in ohms is equal to the time in seconds required for the current to build up to 63.2 per cent of the maximum value. This period is the "inductive-time constant."

It is interesting and significant to note that the time-current relation in an inductive circuit is directly comparable to the time-temperature relation in a coil either heating or cooling. It has been demonstrated that there is a definite time period in which a heating coil will reach half the ultimate temperature rise and that in each successive period it will rise half the remaining balance. The current increases or decreases in exactly the same fashion, for in the first period of time it reaches half the ultimate change in value whether increasing or decreasing. In subsequent equal periods of time, the change is one-half of the remaining possible change. The length of this uniform period of time multiplied by 1.442 is known as the "inductive-time constant" and is the time required for the current to rise or fall 63.2 per cent. Curve Fig. 1 shows these conditions clearly and the relation between "initial time rate" and "inductive-time constant." Expressed as formulas, this takes the following forms:

$$s = L/R$$
$$S = 4 \times L/R$$
$$P = 0.693\,s$$

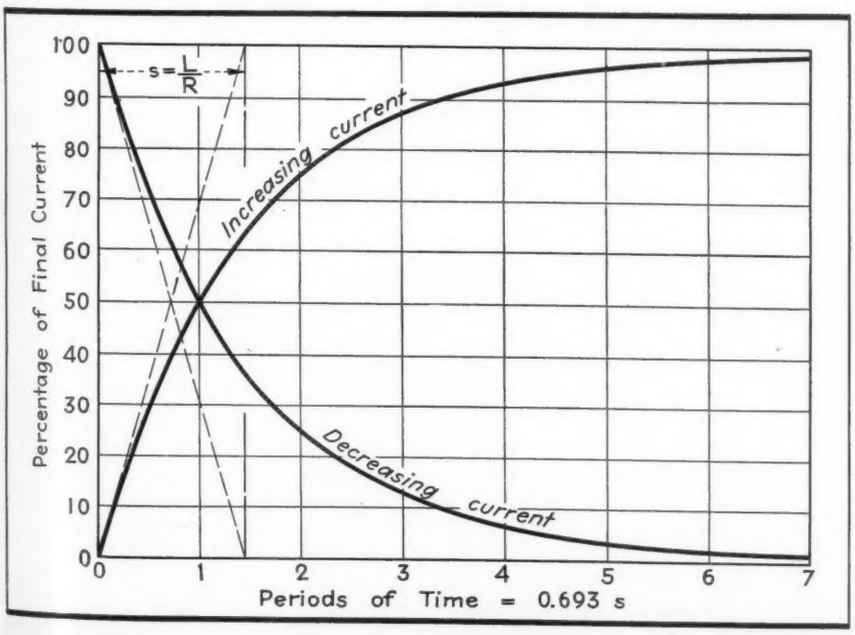

Fig. 1—Relation between current and time in inductive circuits with constant d.c. voltage

Where the following nomenclature is used:

s = Inductive-time constant or the time in seconds required for the current to reach 63.2 per cent of its final value
S = Time in seconds required for the current to reach 98 per cent of its final value
P = Time required for the current to build up from zero to 50 per cent of its ultimate value
L = Inductance in henries
R = Resistance in ohms

On d.c. devices the only interest in inductance is in its damping effect during the first short interval of time or in its effect on circuit interruption. However, on a.c. apparatus the "inductive-time constant" may be longer than the natural period of the applied frequency. This results in the current not having time to build up to its maximum value before the voltage reverses. When this is the case it is the inductance which limits current rather than resistance.

To consider inductance from a different point of view it must be brought out that inductance is an inherent property of coiled conductors. It is the result of the interlinking of magnetic flux of these parallel coiled conductors. Changes in current produce changes in the resultant flux. When the number of lines of flux are changed they induce voltages in the conductors which produce them. These voltages are always in such a direction as to oppose the change producing them. From this it is evident that inductance is an electro-magnetic effect. Therefore the material of the magnetic circuit is an important factor. Coils having no iron possess lower inductance than coils on unsaturated iron circuits. The following formulas are useful for estimating the inductance of coils with air cores:

$$L = \frac{2.03\,a^2\,T^2}{(6a + 9b + 10c)\,10^6}$$

$$L = \frac{0.366\,K\left(\frac{l}{1{,}000}\right)^2}{b + c + r}$$

Where

L = Inductance in henries
T = Number of turns
l = Length of conductor in feet.

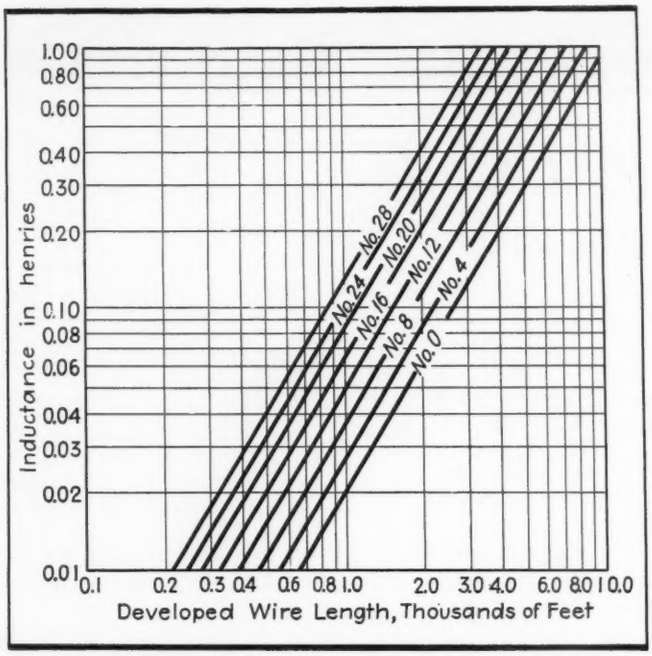

Fig. 2—Curves showing maximum inductance obtainable using various sizes and lengths of DCC wire

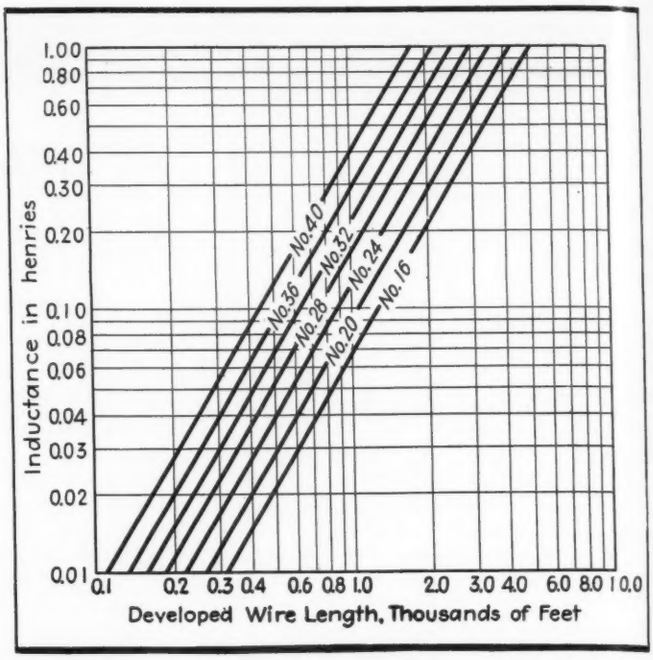

Fig. 3—Curves showing maximum inductance obtainable using various sizes and lengths of enameled wire

K = Shape constant which varies from 1.0 for very long coils to 2.0 for very short coils. For most shapes this falls between 1.1 and 1.2
a = Mean radius of winding
b = Axial length of winding space
c = Depth of winding radially
r = Outside radius of winding

Maximum possible inductance is obtained on coreless cylindrical coils when the proportions are:

$a = 1.5 \qquad b = 1.2 \qquad c = 1.0$

Therefore the maximum inductance ratios of these dimensions are:

$$\frac{a}{b} = \frac{1.5}{1.2} = 1.25$$

$$\frac{b}{c} = \frac{1.2}{1.0} = 1.2$$

Curves Fig. 2 and Fig. 3 show the maximum inductance possible from the best arrangement of air core coil for a given size and length of wire. Changing any of the factors reduces the inductance but the limits are not as sharply critical as might be expected. For example, the inductance will be within 90 per cent of the maximum possible under the following conditions:

$$\frac{a}{b} = 0.5 \text{ to } 2.5, \quad \frac{b}{c} = 1.2$$

$$\frac{a}{b} = 0.7 \text{ to } 1.5, \quad \frac{b}{c} = 2.0$$

Where unsaturated iron is present the inductance will be increased considerably over the case where an air core is used. When the flux can be determined from formulas given in earlier articles of this series on magnetic circuits,

$$L = \frac{\phi \, T}{10^8 \times I}$$

Where

L = Inductance in henries
ϕ = Total flux
T = Number of turns
I = Current in amperes

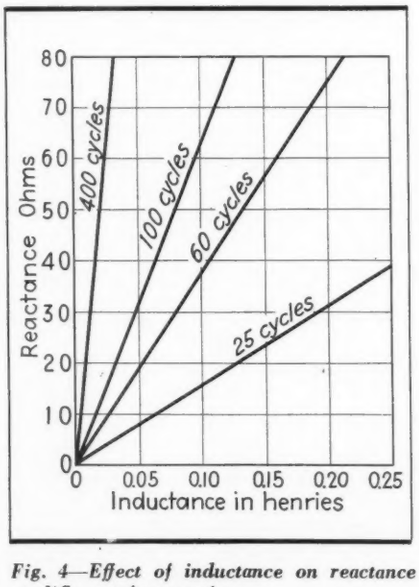

Fig. 4—Effect of inductance on reactance at different frequencies

In general where iron is present in magnetic circuits it is advisable to determine by test the magnetic constants of the device. This can be done either on the exact apparatus or on a similar device and the results interpreted with suitable correction.

The apparent resistance of an inductive device on a.c. circuits can be translated into "reactance ohms" which depend upon frequency and inductance. Curve Fig. 4 illustrates this relation. For calculating the reactance ohms,

$$X = 2 \pi f L$$

Where

X = Reactance ohms
f = Frequency in cycles per second
L = Inductance in henries

Since resistance ohms and reactance ohms are different in their nature they cannot be added directly arithmetically. They must be combined algebraically by the following formula into a combined value known as "impedance ohms."

$$Z = \sqrt{R^2 \times X^2}$$

Where

Z = Impedance ohms
X = Reactance ohms
R = Resistance ohms

On a.c. circuits where the current is limited by a combination of reactance and resistance in the form of impedance ohms, the voltage divided by the impedance ohms is equal to the current in amperes according to Ohm's law.